USAAMRDL-TR-76-8

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HIGH-SURVIVABLE TRANSMISSION SYSTEM

Bell Helicopter Company
P. O. Box 482
Fort Worth, Tex. 76101

May 1976

Final Report for Period January 1974 - August 1975

Approved for public release;
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Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

This report provides a survivable-design approach for a helicopter transmission to allow 60 minutes of operation following the loss of the normal lubrication system. The high-survivable transmission design concept tested under this program incorporated an emergency lubrication system and material improvements to various internal components. This resulted in the modified transmission operating for 4 hours prior to failure.

Results of this contractual effort are still preliminary, and additional effort is required to improve and validate the survivability characteristics of the design.

Mr. Harold Holland of the Military Operations Technology Division served as project engineer for this effort.

Unclassified

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REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER (18) (19) USAAMRDL-TR-76-8	2. GOVT ACCESSION NO.
3. RECIPIENT'S CATALOG NUMBER (9)	
4. TITLE (and Subtitle) (6) HIGH-SURVIVABLE TRANSMISSION SYSTEM	5. TYPE OF REPORT & PERIOD COVERED: Final Report, Jan [redacted] 74-Aug [redacted] 75
	6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) (10) David J. Richardson	8. CONTRACT OR GRANT NUMBER(s) (15) DAAJ02-74-C-0019 NEW
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Bell Helicopter Company, Post Office Box 482, Fort Worth, Texas 76101	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: 62205A IF262205AH88 01 005 EK
11. CONTROLLING OFFICE NAME AND ADDRESS: Eustis Directorate, U.S. Army Air Mobility R&D Laboratory, Ft. Eustis, Virginia 23604	12. REPORT DATE (11) May [redacted]76
	13. NUMBER OF PAGES 113 (12) 114p
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office) (16) DA-1-F-262205-AH-88	15. SECURITY CLASS. (of this report) Unclassified
	15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

~~Approved for public release; distribution~~ unlimited.

(17) 1-F-262205-AH-8801

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Helicopters, Transmission Survivability, Lubrication, Marginal Lubrication, Transmission Thermal Mapping, Gears, Bearings, Transmission Oil Cooler, Transmission Elevated Temperature Operation

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The purpose of this program was to design, fabricate, and test an integrated survivable transmission system for the AH-1G/Q helicopter that would be capable of operation for 60 minutes following the loss of the normal lubrication system. Utilizing the results of previous contractual research efforts, an engineering analysis was performed which produced a design that included an emergency lubrication system in conjunction with some improved components. OVER

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Item 20. (Continued)

One transmission which incorporated the design modifications was fabricated and tested. The test program included an emergency lubrication test run in which a ballistic hit resulting in total loss of the main lubrication system was simulated, and the transmission was run on emergency lubrication at 950 input horsepower (85% of takeoff horsepower) with 25 horsepower through the tail rotor until failure occurred. The transmission ran 4.0 hours following the loss of the normal lubrication system. After 4.0 hours of emergency running, the teeth of the lower sun gear were stripped off, resulting in complete loss of the mast torque and termination of the test.

The design concept for the high-survivable transmission (HST) system tested during this program appears to be more than adequate to provide 60 minutes of transmission operation following the loss of the normal lubrication system. Test results indicate that the transmission could have operated indefinitely if the emergency oil had not leaked past the input seal.

PREFACE

This report contains the results of a program to design, fabricate, and test an integrated survivable main transmission system for the AH-1G/Q helicopter which would be capable of 60 minutes of operation following the loss of the normal lubrication system. This program was conducted by Bell Helicopter Company (BHC) for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory (USAAMRDL) from January 1974 to August 1975 under Contract DAAJ02-74-C-0019.

USAAMRDL technical direction was provided by Harold Holland. The program was conducted under the technical direction of C. E. Braddock, Project Engineer, of the BHC Transmission Design Group. Technical assistance was provided by C. A. Turner, J. H. Drennan, C. L. Baskin, and D. J. Richardson of the BHC Transmission Design Group and R. T. Jenkins, J. D. Rockwood, and P. G. Williams of the BHC Transmission Research Laboratory.

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1.0 INTRODUCTION

Recent combat operations have shown that the helicopter power train is highly susceptible to combat damage due to the remote location of components that serve the main rotor transmission. Ballistic damage can be severe enough to cause loss of lubricant to the rolling elements, such as gears and bearings, and thus limit the time that a pilot has to return to a friendly territory.

Under previous contractual research efforts, many zero lubrication concepts for helicopter transmissions have demonstrated various degrees of survivability after complete or partial loss of conventional lubrication.

The objective of the work performed under this contract was to utilize the results of the previous research efforts, conduct an engineering analysis of the AH-1G/Q helicopter transmission, prepare a design of an ingegrated main transmission, and demonstrate by test that it would be capable of operation for 60 minutes after loss of the regular lubrication system. Design considerations included minimizing modifications to the existing AH-1G/Q helicopter transmission.

Phase I of this contract included analysis and preliminary design. Utilizing the results of previous research efforts involving survivable transmission system concepts, an engineering analysis was conducted to determine the most suitable techniques for continuous lubrication and heat rejection for incorporation in a preliminary design of an integrated survivable transmission system. The analysis included reliability, maintainability, cost, and weight considerations. It was determined that a system including an emergency oil sump with a mechanically driven emergency pump integrated in the main transmission housing showed the most promise in meeting the program objective of 60 minutes of operation after loss of normal lubrication. Component improvements which will be discussed later were also proposed.

Phase II included incorporating the proposed modifications in an existing transmission. Detail and assembly drawings of the integrated trasmission system were prepared and one integrated transmission system was fabricated and assembled.

Phase III was the test program. This included functional tests to determine the transmission response to the emergency lubrication system and the emergency lubrication 60-minute run to determine the time the transmission could be successfully operated at 950 horsepower with the emergency lubrication system operating.

2.0 PRESENT AH-1 TRANSMISSION

Shown in Figure 1 is the present Bell AH-1G type main transmission. It is a typical Bell UH-1 type configuration consisting of a bevel gear set at the main input from the engine supported by a triplex ball bearing and a roller bearing. This input bevel set drives the helicopter mast through two planetary gear trains. The tail rotor drive system is also driven by the input bevel set through a spur gear set and a bevel gear set. This transmission is rated at 1100 horsepower continuous. The lubrication system contains 11 quarts of oil, which are circulated by a 10.5-gallon per minute pump. During normal operation, the oil supply is routed through an oil cooler bypass valve, and then to the transmission as shown schematically in Figure 2. At the transmission, the oil supply enters a manifold and is fed to oil jets for distribution throughout the transmission. The bypass valve is simply a balanced piston device which operates when low pressure exists in the cooler loop. With this system, if the cooler and its associated lines sustain ballistic damage or begin to leak, the bypass valve routes the oil directly to the transmission by sensing a pressure difference and shifting flow. This bypass valve in itself has increased the survivability of the AH-1G helicopter. A ballistic hit taken by the cooler or a line between the cooler and the bypass valve does not result in loss of the regular oil supply. It simply results in the regular oil supply not being cooled. This system is still vulnerable to ballistic damage in the lubrication lines other than the cooler loop or a ballistic hit in the main sump. Either of these would result in complete lubrication loss and, hence, a 7- to 9-minute transmission life based on actual field data and failsafe testing conducted at BHC.

3.0 DESCRIPTION OF SURVIVABILITY PROBLEM AREAS

The first step in designing a transmission capable of 60 minutes of operation following loss of normal lubricant was to define the components of the transmission which severely limit transmission life after lubrication is lost. Previous thermal mapping tests[1] and actual loss-of-oil failures pointed

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[1] Drennan, J. H., and Walker, R. D., Transmission Thermal Mapping (UH-1 Main Rotor Transmission), USAAVLABS Technical Report 73-90, U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, December 1973.

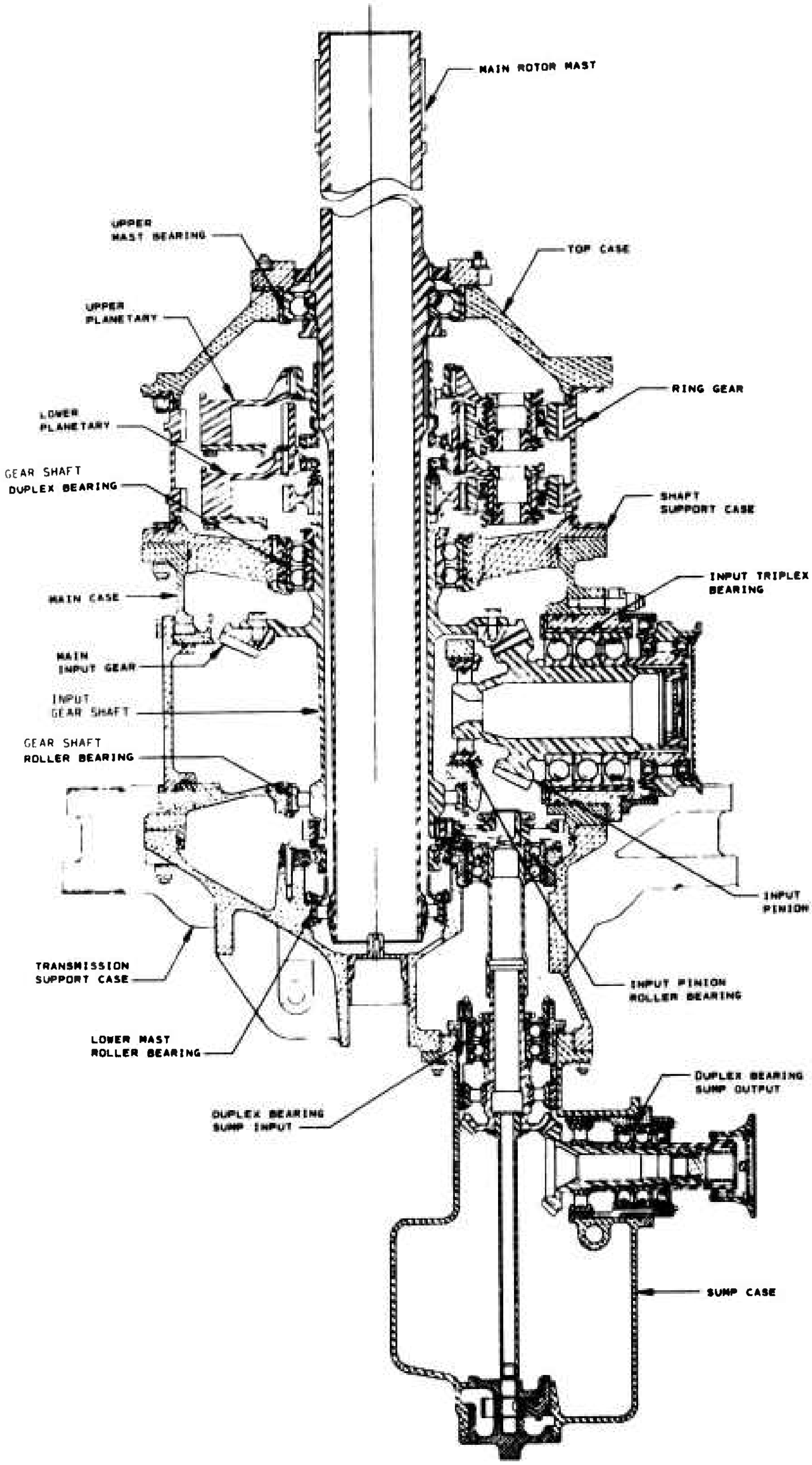

Figure 1. Typical AH-1G/Q type transmission.

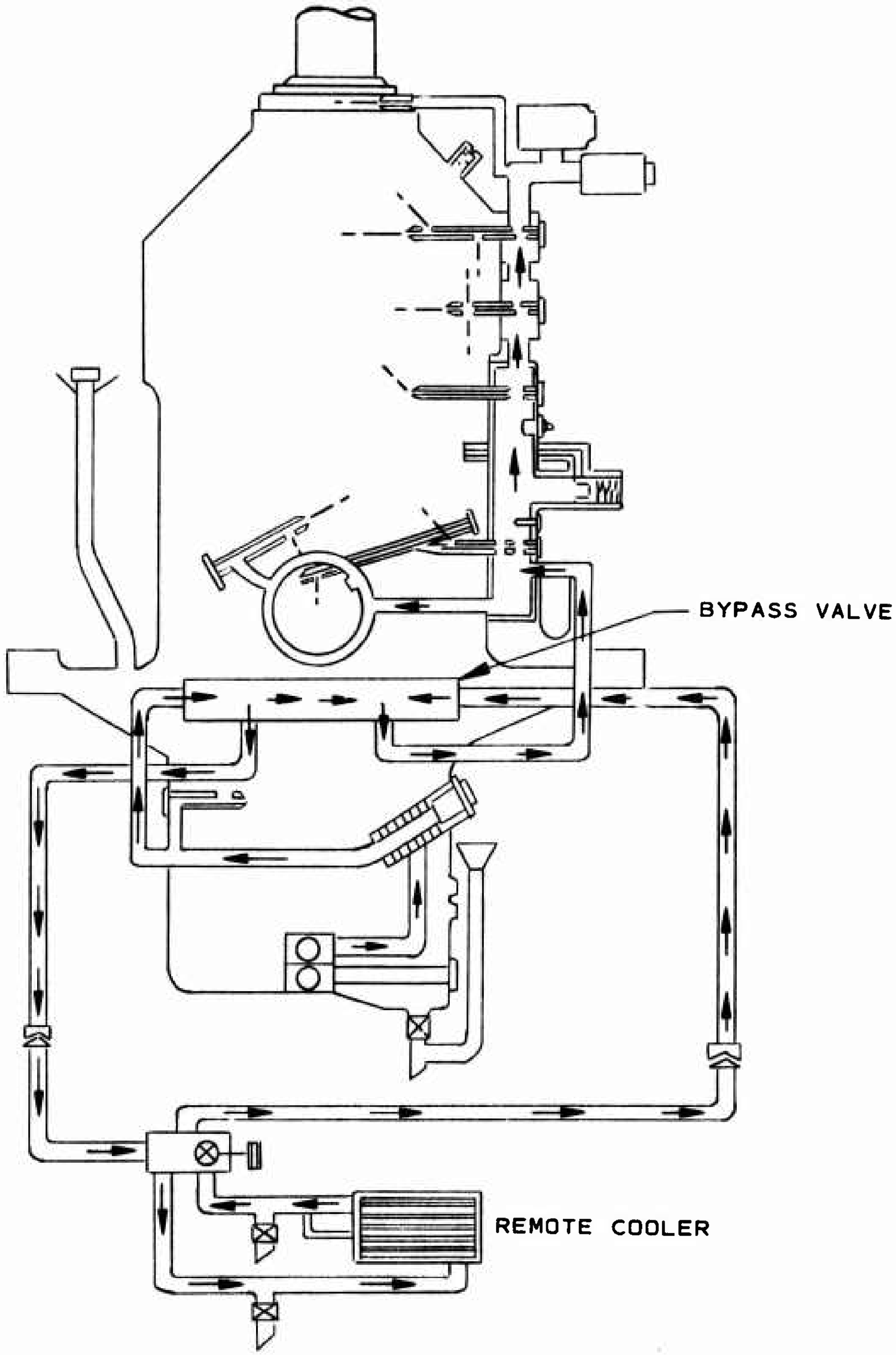

Figure 2. Schematic of present AH-1G/Q lubrication system.

out several problem areas. The main input to the transmission from the engine is a primary problem area due to the high operating speeds and high loads generated in this area. A triplex ball bearing set and a single roller bearing support a bevel pinion in this input quill. Heat generation after loss of lubricant in this area is extremely high, and as presently designed, the life of this quill is only about 7 to 9 minutes following lubrication loss. The mode of failure in this area is usually seizure of the triplex ball bearing due to loss of internal clearance as a result of thermal expansion of the bearing components or failure of the bevel pinion teeth due to loss of backlash as a result of thermal expansion.

Previous fail-safe tests performed at BHC have indicated that the planetary assemblies are also a major life-limiting area following loss of lubrication. The lower planetary assembly contains bronze roller bearing retainers which have low strength in the high-temperature range of marginal lubrication or dry operation. During two of the fail-safe test runs, failures after 6 minutes and after 6 minutes 20 seconds of nonlubricated running occurred due to the bronze roller bearing retainer transferring material to bearing races and rollers which ultimately resulted in breakage of the tangs which retain the rollers. The upper planetary assembly contains nylon roller bearing retainers which are severely limited by the melting point of the nylon. After 12 minutes of nonlubricated operation during fail-safe testing, failure occurred due to the nylon retainers melting and flowing throughout the assembly. These tests also showed that loss-of-lubricant operation was limited by the loss of hardness of the 52100 steel rollers and the carburized thrust washers of the planetary assemblies. During dry operation, loss of clearance in the sun gear-planet pinion mesh due to thermal expansion can also occur and result in transmission failure.

These were the main areas of component improvement upon which the redesign focused.

4.0 DESCRIPTION OF HST SYSTEM

The transmission used for this program was a UH-1D type (part number 205-040-001-5, serial number A12-52) modified to an AH-1G configuration and further modified with the HST components as described below. Figure 3 depicts the modified transmission system, and Table 1 includes a description of all parts modified. Table 2 lists the new parts fabricated. Survivability enhancement was accomplished through two major categories of modification. These were component improvements and installation of an emergency lubrication system.

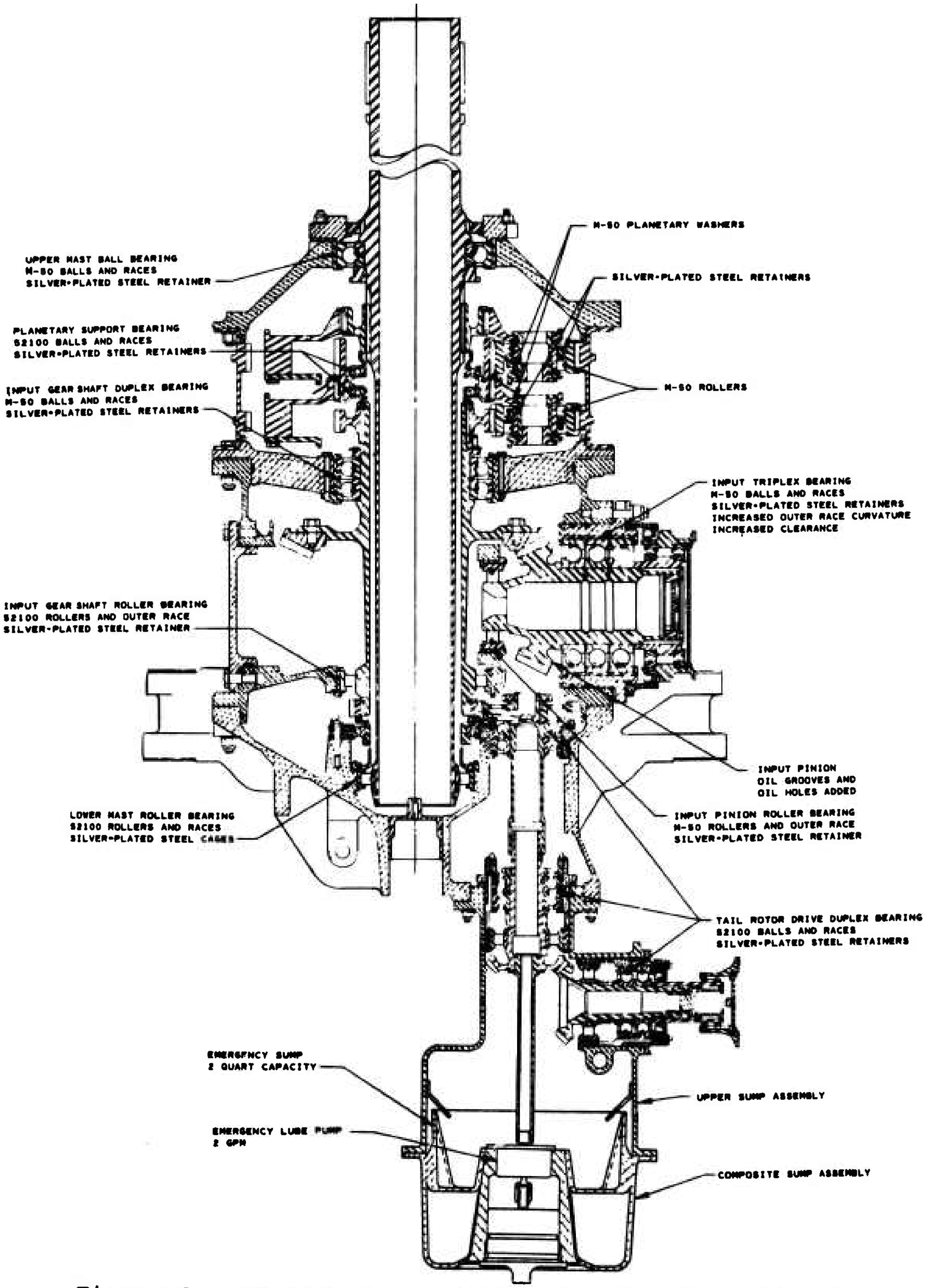

Figure 3. AH-1G/Q transmission with high-survivable modifications.

TABLE 1. PARTS MODIFIED FOR USE IN THE HST SYSTEM

Part name	HST P/N	P/N replaced	Description of modification
Upper Planetary Retainers	300-040-168-3	204-040-129-1	Changed material from nylon to silver-plated steel.
Lower Planetary Retainers	300-040-168-3	204-040-133-1	Changed material from bronze to silver-plated steel.
Planetary Washers	299AWH-03-1	204-040-134-5	Changed material from AMS6260 steel to M-50 steel.
Planetary Support and Tail Rotor Takeoff Ball Bearing	299AWH-18-1	204-040-135-1	Retainer material changed from bronze to silver-plated steel.
Upper Mast Ball Bearing	299AWH-02-7	204-040-136-9	Retainer material changed from bronze to silver-plated steel.
Tail Rotor Drive Duplex Ball Bearing	299AWH-02-5	204-040-143-1	Retainer material changed from bronze to silver-plated steel.
Lower Mast Roller Bearing	299AWH-18-3	204-040-270-1	Retainer material changed from bronze to silver-plated steel.
Lower Gearshaft Roller Bearing	299AWH-18-5	204-040-271-3	Retainer material changed from bronze to silver-plated steel.
Planetary Roller Sets	204-040-725-5	204-040-725-3	Material changed from 52100 steel to M-50 steel.
Input Gearshaft Duplex Bearing	299AWH-02-3	205-040-245-1	Retainer material changed from nylatron to silver-plated steel.
Input Triplex Bearing	299AWH-02-1	205-040-246-1	Outer race curvature changed from 52% to 54% and oil grooves added to inner races.
Main Input Pinion Roller Bearing	299AWH-13-1	205-040-249-1	Retainer material changed from bronze to silver-plated steel.
Oil Pump Shaft	299AWH-19-1	204-040-197-3	Shortened to accommodate emergency lube pump.
Oil Transfer Tube	299-040-250-3	204-040-223-11	Modified to accommodate emergency system plumbing.
Main Oil Pump Screen	299AWH-28-1	204-040-237-1	Shortened to fit in new composite sump assembly.
Oil Jet	299AWH-08-1	204-040-326-7	Modified to accommodate emergency system plumbing.
Oil Jet	299AWH-12-1	204-040-327-1	Modified to accommodate emergency system plumbing.
Oil Housing	299AWH-14-1	204-040-339-5	Modified to accommodate emergency system plumbing.
Transmission Main Case	299HES-997-1	204-040-353-23	Modified to accommodate emergency system plumbing.
Sump Case	299HES-1004-1	204-040-355-5	Modified to accommodate new lower composite sump assembly.
Oil Manifold	299-040-204-1	204-040-393-1	Modified to accommodate emergency system plumbing.
Oil Fitting	299-040-205-1	204-040-398-1	Modified to accommodate emergency system plumbing.
Main Input Pinion	299AWH-06-1	204-040-700-1	Oil grooves and oil holes added to supply triplex from pinion interior.

TABLE 2. NEW PARTS REQUIRED FOR EMERGENCY LUBRICATION SYSTEM

Part name	HST P/N	Description
Oil Manifold	299-040-204-3	Internal manifold for emergency system plumbing.
Elbow	299-040-204-5	Internal elbow for emergency system plumbing.
Oil Transfer Tube	299-040-204-7	Transfer tube for emergency system plumbing.
Oil Manifold	299-040-205-5	Internal manifold for emergency system plumbing.
Tube	299-040-206-1	Internal plumbing for emergency system.
Tube	299-040-207-1	Internal plumbing for emergency system.
Tube	299-040-208-1	Internal plumbing for emergency system.
Composite Sump Assembly	299-040-217-1	Emergency sump within a main sump. Manufactured from Hexcel F-161 epoxy and scrim cloth.
Oil Transfer Tube	299AWH-16-1	Transfer tube for emergency system plumbing.
Emergency Oil Pump Screen	299AWH-23-1	Inlet screen for emergency oil pump.
Emergency Lubrication Pump	SKC 2750	Oil pump for emergency lubrication system.
Check Valve	C2495-1Q	Check valve for main lube system/emergency lube system interface.
Relief Valve	532-3M-20	Relief valve for regulation of emergency lubrication system.

4.1 COMPONENT IMPROVEMENTS

In the input quill area both the triplex ball bearing and the roller bearing were modified. The bronze retainer of the roller bearing was replaced with a silver-plated steel retainer to afford better high-temperature operation capabilities. The modified roller bearing is shown in Figure 4. Figure 5 shows the modified triplex bearing. The outer race curvature of the input triplex ball bearing was increased from 52% to 54% of the ball diameter in an effort to transfer the bearing race control from the outer race to the inner race. Thus, ball sliding should occur at the outer race of the modified triplex with pure rolling occurring at the inner race. The ultimate goal of this modification was to increase heat generation at the outer race while descreasing heat generation at the inner race correspondingly. This was desirable since failure of the standard input triplex bearing could occur due to loss of bearing clearance when the inner race became much hotter than the outer race. Thermal mapping tests of a UH-1D type transmission had shown that in normal operation the inner race of the standard input triplex bearing ran significantly hotter than the outer race. The race curvature change of the modified triplex tended to equalize the temperatures of the inner and outer races, and, indeed, in the testing performed during this program the outer race was always hotter than the inner race. This curvature change also resulted in a .003-inch increase in internal clearance in the modified triplex bearing.

The input pinion was modified by adding two oil grooves and six oil holes to allow oil to flow from the pinion interior to the input triplex bearing for emergency lubrication operation. The modified pinion is shown in Figure 6.

In the planetary assemblies shown in Figure 7 and Figure 8, several material changes were made based on previous fail-safe testing of the planetaries. These changes were effected to achieve better high-temperature strength characteristics of these components. The bronze bearing retainers of the lower planetary assembly and the nylon bearing retainers of the upper planetary assembly were replaced with silver plated steel retainers. Figure 9 shows the three types of retainers. Planetary thrust washers manufactured from CEVM M-50 steel replaced the AMS 6260 steel thrust washers. The A1S1 52100 steel planetary rollers were replaced with CEVM M-50 steel rollers.

Table 1 lists the other bearings in the transmission which were modified. In the modified transmission tested, only four bearings did not have silver-plated steel retainers. The two ball bearings in the freewheeling unit had bronze retainers. These bearings do not turn unless the transmission overruns the

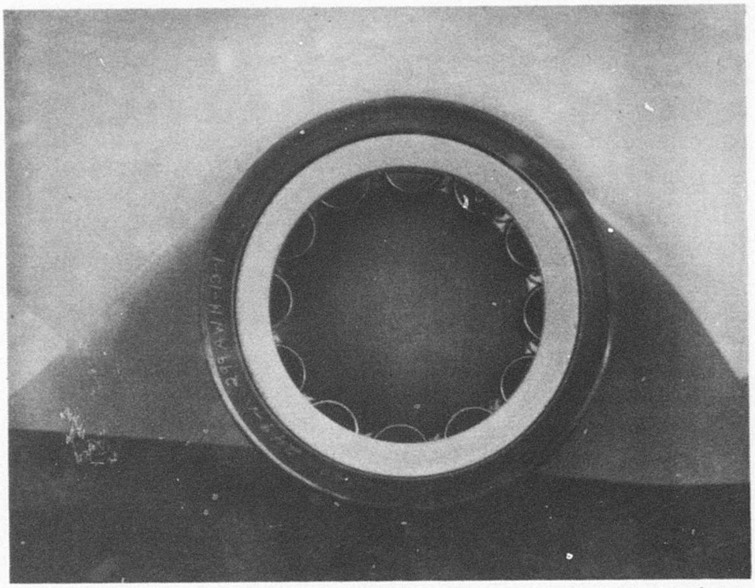

Figure 4. Input pinion roller bearing with silver-plated steel retainer.

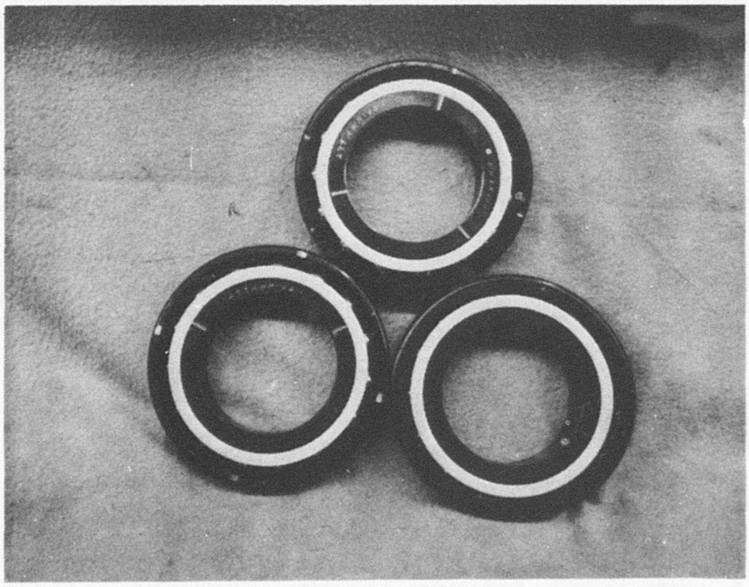

Figure 5. Input triplex bearing with increased outer race curvature.

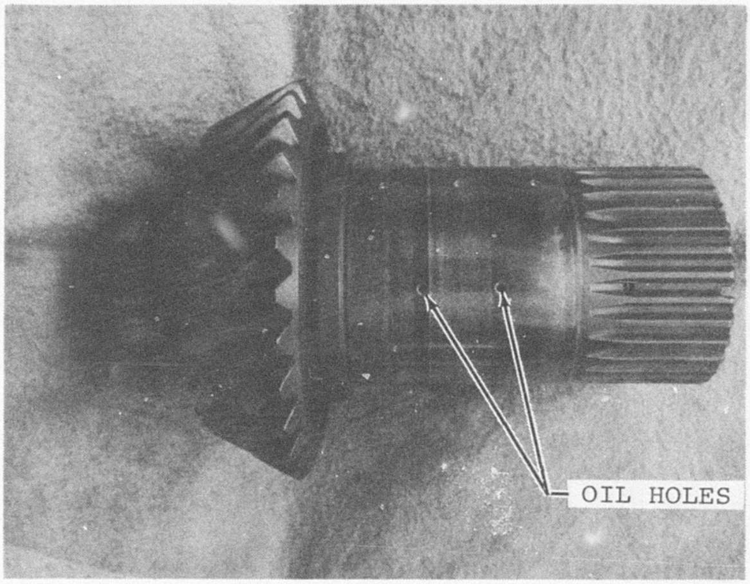

Figure 6. Main input pinion with additional oil holes.

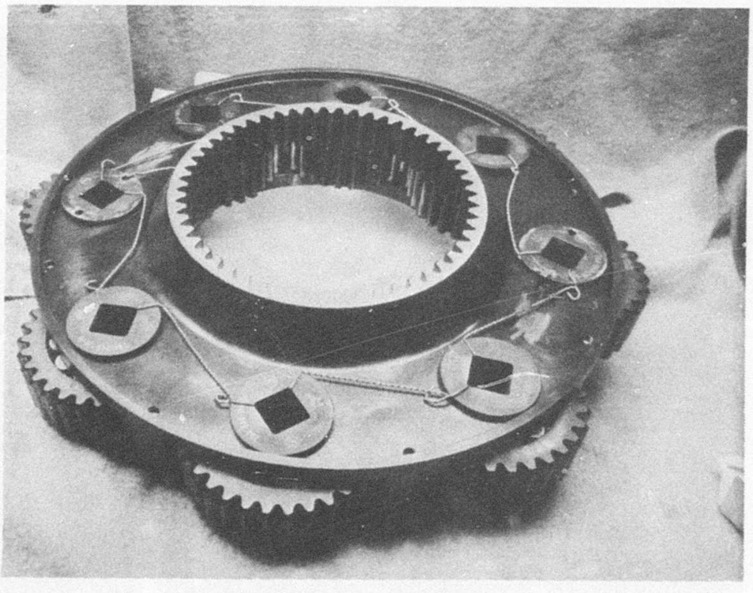

Figure 7. Upper planetary assembly with HST modifications.

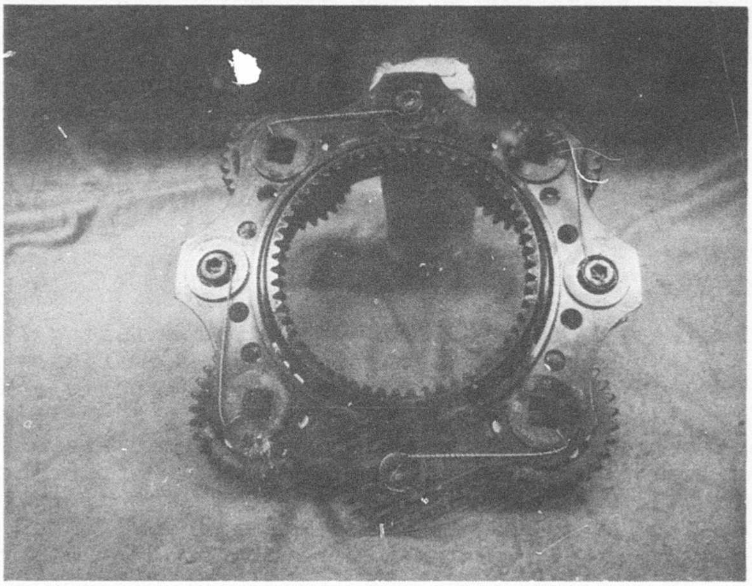

Figure 8. Lower planetary assembly with HST modifications.

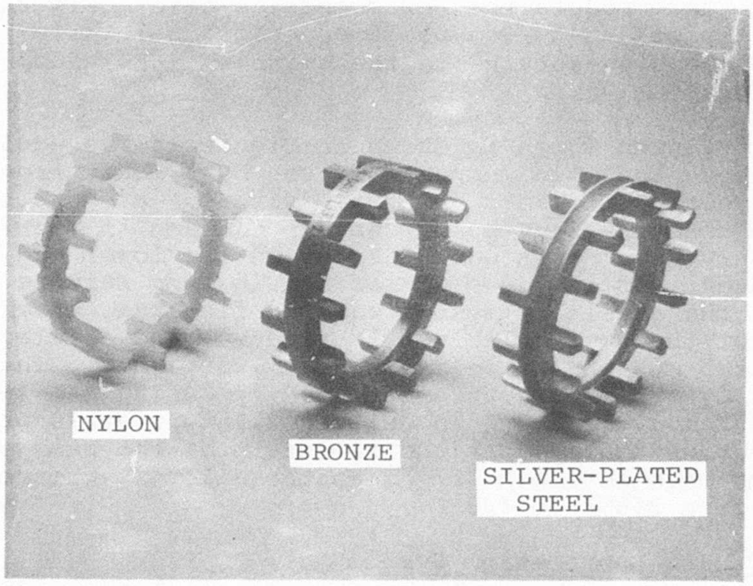

Figure 9. Nylon, bronze, and silver-plated steel planetary roller bearing retainers.

engine, and thus there was no need to modify them. The two 204-040-310 roller bearings of the tail rotor and accessory drive bevel gear set also had bronze retainers. Because these bearings are roller bearings (roller bearings usually generate less heat than ball bearings) and because of the remoteness of these bearings relative to the predicted hot spots of the transmission, these bearings did not require modification.

4.2 EMERGENCY LUBRICATION SYSTEM

4.2.1 General Description

The second category of modifications to the existing AH-1G transmission involved the installation of an emergency lubrication system. This system utilized a continuously running mechanically driven pump to deliver lubricant to critical components in the transmission. The pump obtained its oil supply from an auxiliary sump located inside the main transmission sump. The auxiliary sump was supported by webs attached to the lower half of a split sump case and was positioned to receive practically all of the oil returning from the transmission. When the auxiliary sump was full, the oil would overflow into the main sump. The usable capacity of the auxiliary sump was approximately two quarts. Figure 10 shows the composite sump case, and Figure 11 shows the upper sump case which was modified to accommodate the new composite case. Figure 12 shows the typical AH-1G/Q-type lower sump case, while Figure 13 depicts the high-survivable transmission (HST) sump assembly configuration.

The auxiliary pump which is shown in Figure 14 was positioned directly above the main pump and driven by a shaft from the sump bevel gear. The main pump was driven by a short splined shaft from the auxiliary pump. As can be seen in Figure 15, the auxiliary pump shaft was undercut on the lower end so that it would shear off in the event of a main pump seizure. Since the oil in the auxiliary system was not filtered, a screen assembly for the pump inlet was fabricated, and an electric chip detector was installed in the auxiliary sump. The pump was tested by the manufacturer at 4500 rpm using MIL-L-23699 oil at 250°F and was found to deliver 2.75 gpm at 20 psi. The manufacturer estimated that the flow rate of the pump could decrease as much as one-half gpm as the oil temperature increased from 250°F to 450°F.

4.2.2 HST Main Lubrication System

A schematic of the lubrication system for the HST system is shown in Figure 16. In normal operation, oil is pumped by the main oil pump through a screen filter to a bypass valve.

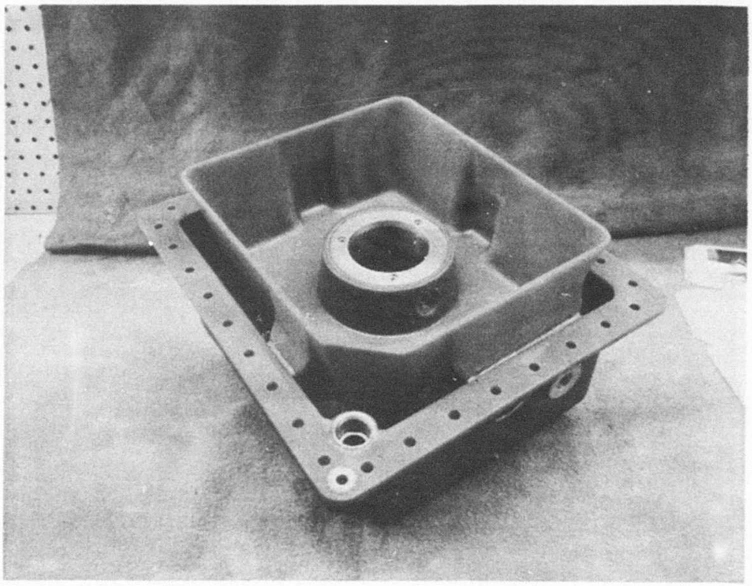

Figure 10. Composite sump case for HST.

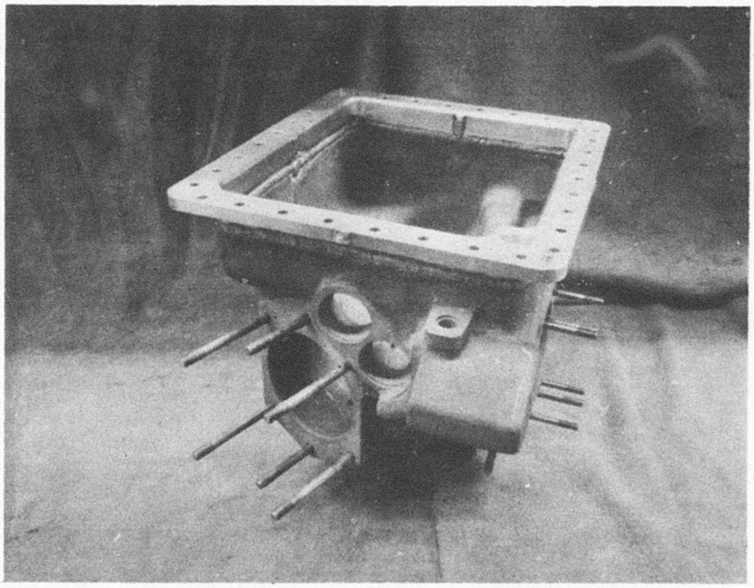

Figure 11. Upper sump case for HST.

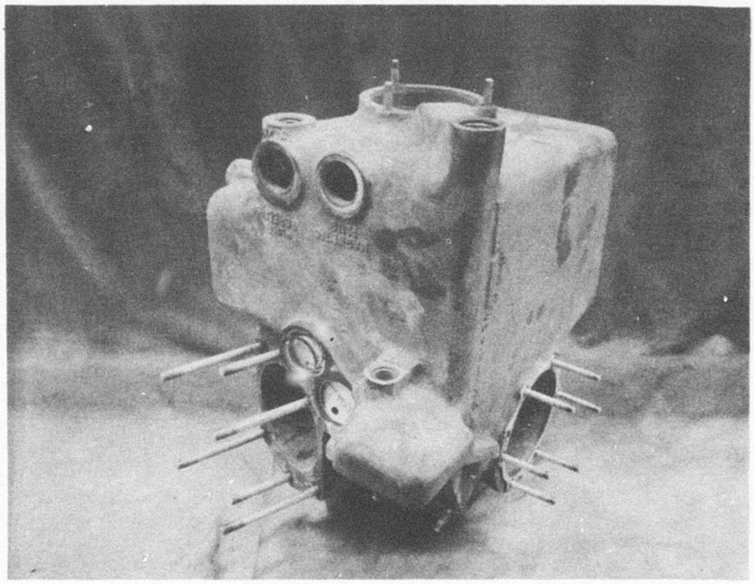

Figure 12. Typical AH-1G/Q type sump case.

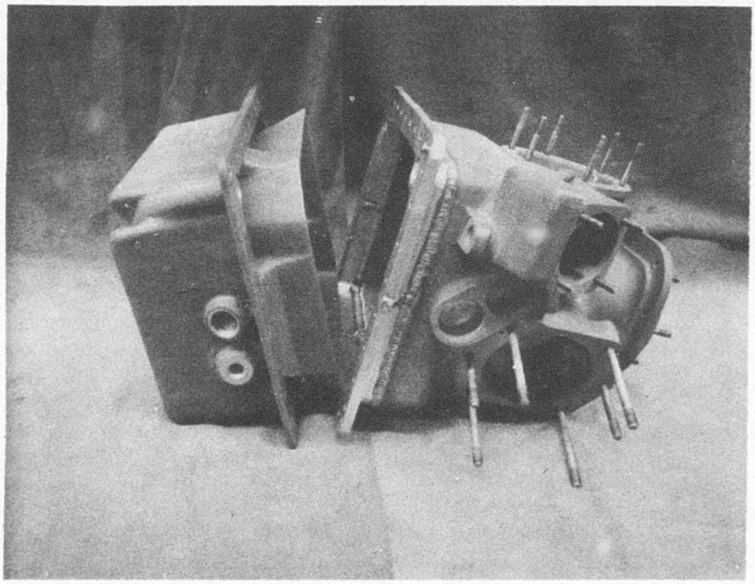

Figure 13. Sump assembly for HST.

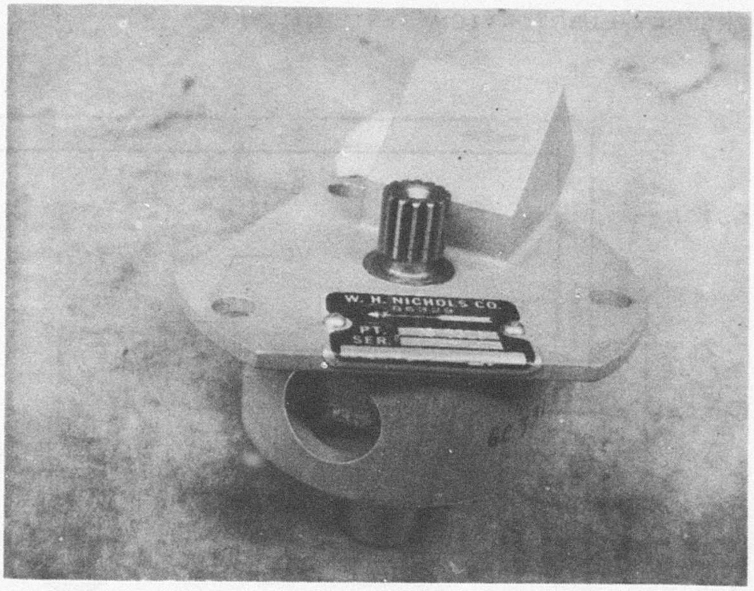

Figure 14. Emergency lubrication pump for HST.

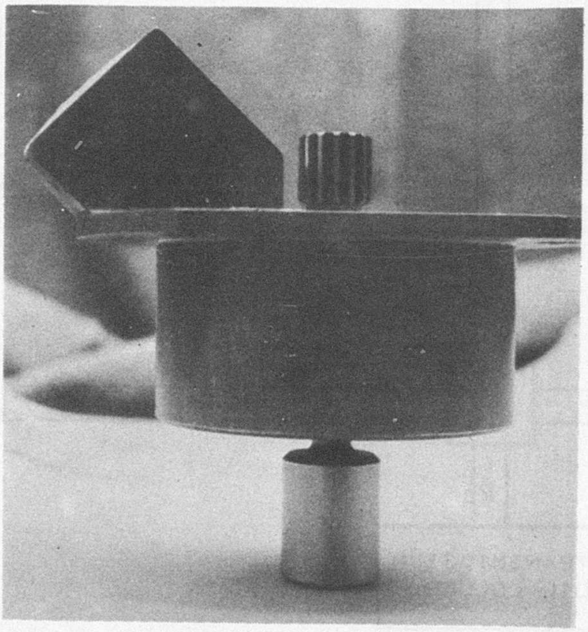

Figure 15. Emergency lubrication pump for HST showing undercut shaft.

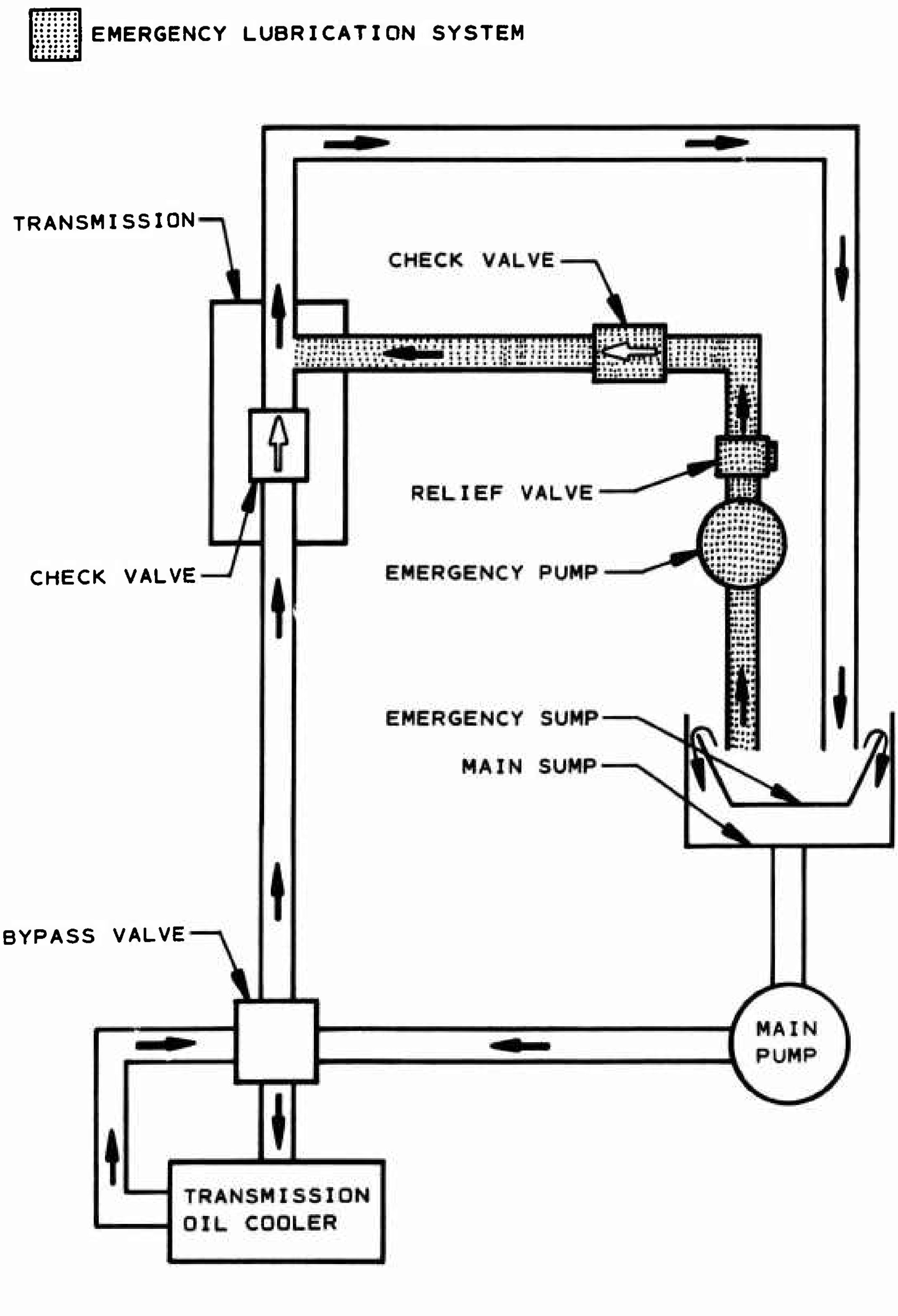

Figure 16. Schematic of lubrication system for HST.

From the bypass valve it is routed through the cooler and then through a filter and back to the transmission. It enters the transmission by way of a manifold that feeds jets for the input bevel gears and bearings. The oil is routed from the manifold through external plumbing to the jets for the planetary assemblies. From here, the oil is routed in two directions. One oil line carries oil to the jets which lubricate the upper mast bearing and the driving spline. The other oil line is an internal line which carries oil to an internal manifold which feeds jets that supply oil to the input bevel pinion roller bearing, to the bevel mesh, and to the interior of the input bevel pinion. The oil inside the pinion is distributed through holes to the input triplex and also out the end of the pinion to the freewheeling unit.

4.2.3 HST Emergency Lubrication System

When oil is lost from the main sump due to ballistic damage or some other cause, two quarts of usable oil are retained in the emergency sump. When the main oil pressure drops below 20 psi, the emergency system becomes fully operative. In emergency operation, oil flows from the emergency oil pump through internal lines to oil jets which supply lubricant to the main input bevel mesh, to the main input pinion roller bearing, and through the interior of the main input pinion to the input triplex bearing and the freewheeling unit. The internal lines then become integral with the upper portion of the main oil supply system and provide lubricant for the planetary assemblies and upper mast bearing. A check valve installed in an external transfer tube adjacent to the ring gear case separates the main oil supply system and the emergency oil supply system. The emergency system is almost all internal, and, therefore, its inherent vulnerability is low. Figure 17 shows some of the internal plumbing installed in the main center case for the emergency lubrication system.

5.0 HST TESTING

5.1 GENERAL OBJECTIVES

Following the complete fabrication and assembly of the high-survivable transmission system, tests were performed to determine the response of the transmission to the following conditions:

- Loss of full effectiveness of the oil cooler. (This could be due to loss of the blower.)

- Total loss of the oil cooler which would result in operating on full bypass.

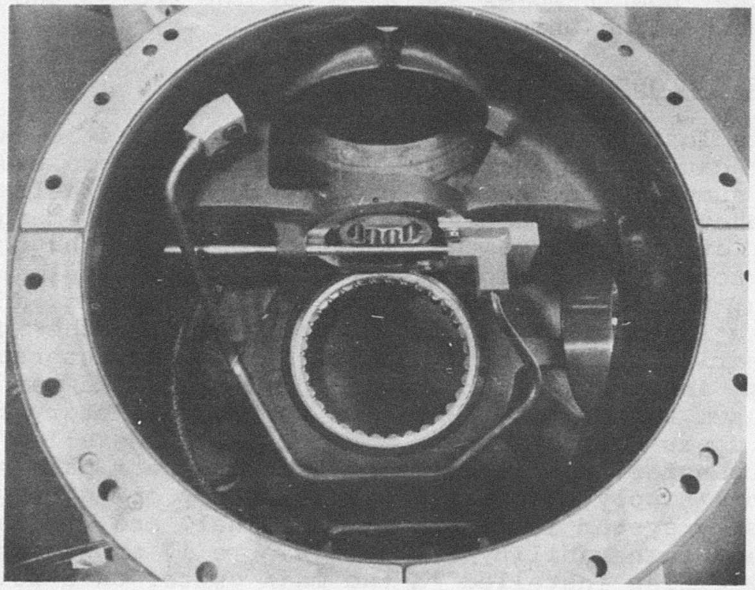

Figure 17. Main center case with internal plumbing for HST.

- Loss of the bypass valve and/or the lower part of the sump which would result in the loss of the main oil supply. The emergency lubrication system would then take over.

- Loss of the emergency oil supply.

The testing was conducted in three phases which are described below.

5.2 PHASE I TESTS

5.2.1 Phase I Test Configuration

For the initial phase of testing, the HST was assembled with a mast which was cut off just below the spline which mates with the upper planetary carrier and was supported in the transmission by a special tandem bearing. This was done to enable thermocouples to be installed to monitor the area in and around the bevel gear shaft. It was desirable to use a standard mast for the Phase II and Phase III testing in order to investigate the effects of emergency lubrication running on the standard mast ball bearing and the mast roller bearing which were modified for the HST program. In order to investigate these effects and yet not lose track of the thermal conditions in and around the bevel gear shaft during the emergency lubrication testing, the stubbed mast with all of the associated thermocouples was used in Phase I to establish a thermal baseline from which a temperature gradient across a bearing could be utilized to estimate the temperature of the inner ring during the Phase II and Phase III testing. The thermocouple locations for the Phase I testing paralleled previous thermal mapping tests and are defined in Figure 18 and Table 3.

The transmission was installed in the test stand. This test stand was a combination regenerative/absorption-type stand. Torque was applied to the main rotor mast through a regenerative loop by rotating the "slave transmission" planetary ring gear in relation to the main case. Main rotor mast torque was monitored by employing a strain-gaged mast. The tail rotor loop was loaded using a 300-horsepower water brake dynamometer. Power was supplied to the system by a 500-horsepower electric motor, and speed was regulated by a magnetic coupling.

A system to simulate a severed oil line was designed and implemented. Provisions for bypassing the oil cooler were also included. The system is shown schematically in Figure 19. Oil from the main oil sump could be pumped through any of three paths: (1) by closing valves A and C and opening valve B, the oil could be pumped through the oil cooler and back to the transmission; (2) by closing valves B and C and opening valve A, the oil could be pumped directly back to the transmission

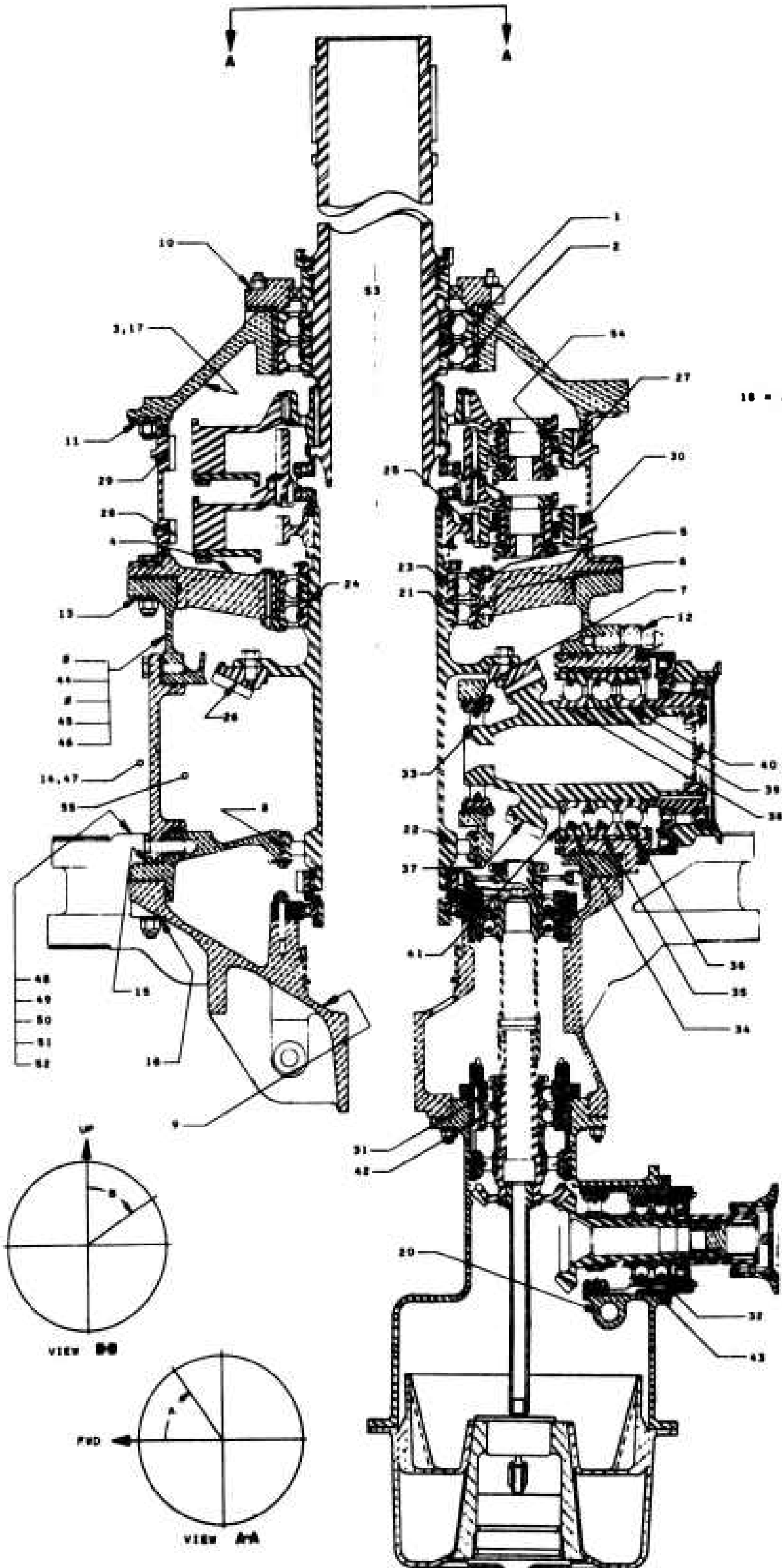

Figure 18. Thermocouple locations for Phase I testing.

TABLE 3. THERMOCOUPLE LOCATION INDEX

Thermocouple Number	Location/Nomenclature	∡A	∡B
1	Mast Bearing Duplex (Upper)	270°	
2	Mast Bearing Duplex (Lower)	300°	
3	Top Case (Inner Surface)	190°	
4	Support Case, Bevel Gear (Upper Surface)	190°	
5	Bevel Gear Shaft Upper Bearing (Outer Ring)	270°	
6	Bevel Gear Shaft Lower Bearing (Outer Ring)	270°	
7	Input Pinion Roller Bearing (Outer Ring)		30°
8	Bevel Gear Shaft Roller (Outer Ring)	120°	
9	Support Case, Main Transmission (Floor)	0°	
10	Mast Adapter Plate	185°	
11	Ring Gear Case Upper Flange	180°	
12	Input Pinoin Quill Sleeve	190°	270°
13	Main Case Upper Flange	280°	
14	Main Case Accessory Port Cover	275°	
15	Main Case Lower Flange	280°	
16	Support Case, Upper Flange, Main Transmission	350°	
17	Top Case (Inner Surface)	10°	
18	Test Cell Ambient Air (12 Inches From Main Case)		
19	Oil-In Temperature Probe	120	

TABLE 3. - Continued

Thermocouple Number	Location/Nomenclature	∡A	∡B
20	Oil-Out Temperature Probe	270	
21	Bevel Gear Shaft Lower Duplex Bearing (Inner Ring)		
22	Bevel Gear Shaft Roller Bearing (Inner Ring)		
23	Bevel Gear Shaft Upper Bearing (Inner Ring)		
24	Bevel Gear Shaft Between Duplex Bearing (Shaft)		
25	Lower Sun Gear Root		
26	Bevel Gear Tooth (Root)		
27	Ring Gear Upper Mesh (Tooth Implant)	45°	
28	Ring Gear, Lower Mesh (Tooth Implant)	45°	
29	Ring Gear, Upper Mesh (Tooth Root)	40°	
30	Ring Gear, Lower Mesh (Tooth Root)	40°	
31	Tail Rotor Drive Input Duplex, Upper Bearing (Outer Ring)	280°	
32	Tail Rotor Drive Output Duplex, Outboard Bearing (Outer Ring)		50°
33	Input Pinion Nose Bearing (Inner Ring)		
34	Input Pinion Inboard Triplex Bearing (Outer Ring)		250°
35	Input Pinion Center Triplex Bearing (Outer Ring)		80°
36	Input Pinion Outboard Triplex Bearing (Outer Ring)		85°

TABLE 3. - Continued

Thermocouple Number	Location/Nomenclature	∡A	∡B
37	Input Pinion Tooth (Root)		
38	Input Pinion Inboard Triplex Bearing (Inner Ring)		
39	Input Pinion Center Triplex Bearing (Inner Ring)		
40	Input Pinion Outboard Triplex Bearing (Inner Ring)		
41	Triplex Bearing, Oil-Out		
42	Tail Rotor Drive Input Duplex Lower Bearing (Outer Ring)	280°	
43	Tail Rotor Drive Output Duplex Inboard Bearing (Outer Ring)		
44	Main Case	95°	
45	Main Case	75°	
46	Main Case	65°	
47	Main Case, Center Section	85°	
48	Support Case, Main Transmission	85°	
49	Support Case, Main Transmission	85°	
50	Support Case, Main Transmission	85°	
51	Support Case, Main Transmission	95°	
52	Support Case, Main Transmission	100°	
53	Air Inside of Output Mast		
54	Upper Planet Pinion Bearing (Inner Ring)		
55	Air/Oil Inside Main Case		

TABLE 3. - Concluded

Thermocouple Number	Location/Nomenclature	$\measuredangle A$	$\measuredangle B$
56	Mast Roller Bearing (Outer Ring)	270°	
57	Mast Ball Bearing (Outer Ring)	0°	

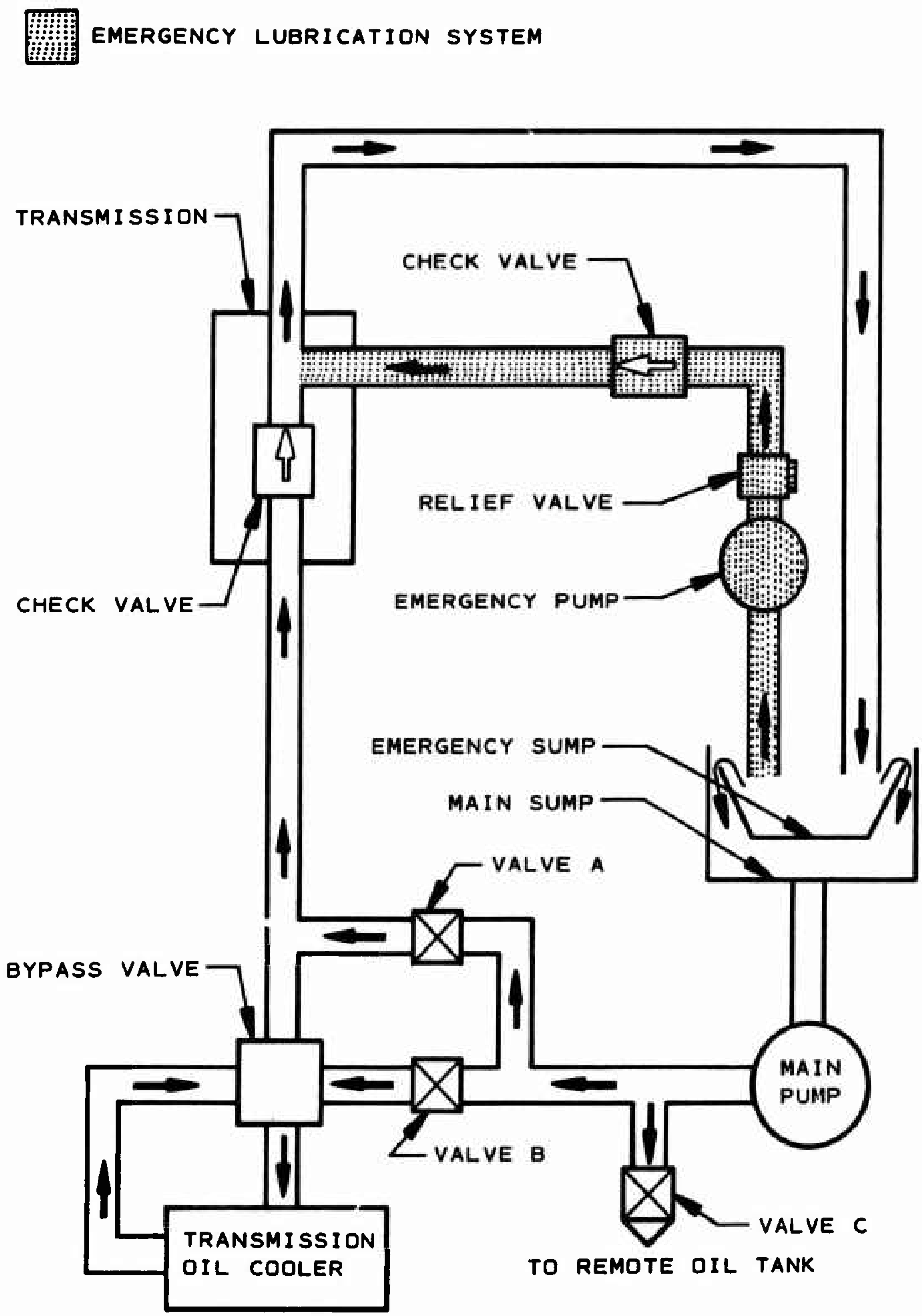

Figure 19. Schematic of HST test configuration to simulate ballistic hit.

without going through the cooler; (3) by closing valves A and B and opening valve C, the oil could be pumped into a remote oil tank, preventing its return to the transmission. This system was functional for all three phases of testing.

Throughout all phases of testing, temperatures were recorded on two Honeywell Electronic 16, 24-point strip chart recorders and one United Systems Corporation Datalogger. The two Honeywell Electronic 16 recorders were type K (alumel-chromel) calibration with a full-scale range of -200°F to +1000°F. The Datalogger was of the type J (iron-constantan) calibration and had a full-scale range of -320°F to +1400°F.

Temperatures were recorded at intervals of 2 minutes or less throughout the testing. Pressures, oil flow, main rotor mast torque, tail rotor mast torque, and input rpm were recorded at intervals of 6 minutes or less. Prior to testing, the transmission and all associated oil lines were flushed and cleaned. The transmission was serviced with MIL-L-23699B lubricant, which was the type of lubricant used throughout all phases of testing.

5.2.2 Phase I Test Objectives

The transmission was test run as described in Table 4 to meet the following specific objectives:

- To verify the functional capability of all new, used, and modified components and to give these parts a break-in period.

- To check out all instrumentation, gages, and flowmeters.

- To obtain a thermal baseline with all thermocouples except one functioning. The one exception was the thermocouple on the inner ring of the outboard bearing of the input triplex. Five thermocouples were required in the input area, but the input drive shaft slip ring assembly would accommodate only four at a time. The thermal baseline for the outboard bearing was obtained during Phase II tests.

5.2.3 Results of Phase I Tests

Only about one-half of Phase I-1 (Build-up Verification and Run-In) was completed since all of the gears were used and already broken in, and the bearings had polished races and balls which reduced the effects of a run-in. Phase I-2 (Thermal Baseline Runs) was completed except for the two 350°F stabilized oil-in temperature runs. Such an extensive length of time was required to reach the 350°F stabilized temperature that it was decided to delete these two runs and proceed with Phase I-3.

The stabilized temperatures recorded for steps 1, 2, and 3 of Phase I-2 are given in Figures 20, 21, and 22, respectively. It is significant to note that the outer rings of the input triplex bearing set were hotter than the inner rings. With the previous design, the inner rings of this bearing maintained a higher temperature than the outer rings (USAAMRDL Technical Report 73-90). Another interesting result was that as the stabilized oil-in temperature was allowed to increase from 200°F to 300°F, the temperature rise (ΔT) across the transmission decreased from 11°F to 4°F, respectively, as can be seen in Figure 23.

Step 3 of Phase I-3 was aborted after about five minutes because when the main oil supply was drained from the main sump, the emergency pump did not supply pressure to the system. After shutting down, the emergency sump was drained and found to contain only about .17 quart. Step 3 was started once again and stopped after one minute with still no indication of oil pressure. The emergency sump was again drained and found to contain about 1.46 quarts. The transmission was then removed from the cell and partially disassembled. An inspection revealed that the check valve directly above the pressure regulator valve near the emergency lube pump was installed backward. Thus, none of the emergency oil was reaching the lube distribution system,which resulted in the lack of pressure. However, as the oil was being dumped through the pressure regulator valve, it still should have been collected in the emergency sump. Since the emergency sump was losing its oil at the rate of approximately one-half quart per minute, it was concluded that a large percentage of the returning oil must have been clinging to the walls of the casting and bypassing the emergency sump. An aluminum skirt was bonded to the upper sump case to direct all of the returning oil into the emergency sump.

5.3 PHASE II TESTS

5.3.1 Phase II Test Configuration

For the second phase of testing, the HST was assembled and instrumented exactly the same as for the Phase I tests with the following exceptions:

- A standard mast with a standard top case was installed.

- The slip ring and thermocouples used to monitor the temperatures in and around the bevel gear shaft were omitted.

- The thermocouple on the inner ring of the main input pinion roller bearing was deleted, and the thermocouple on the inner ring of the outboard triplex bearing was activated.

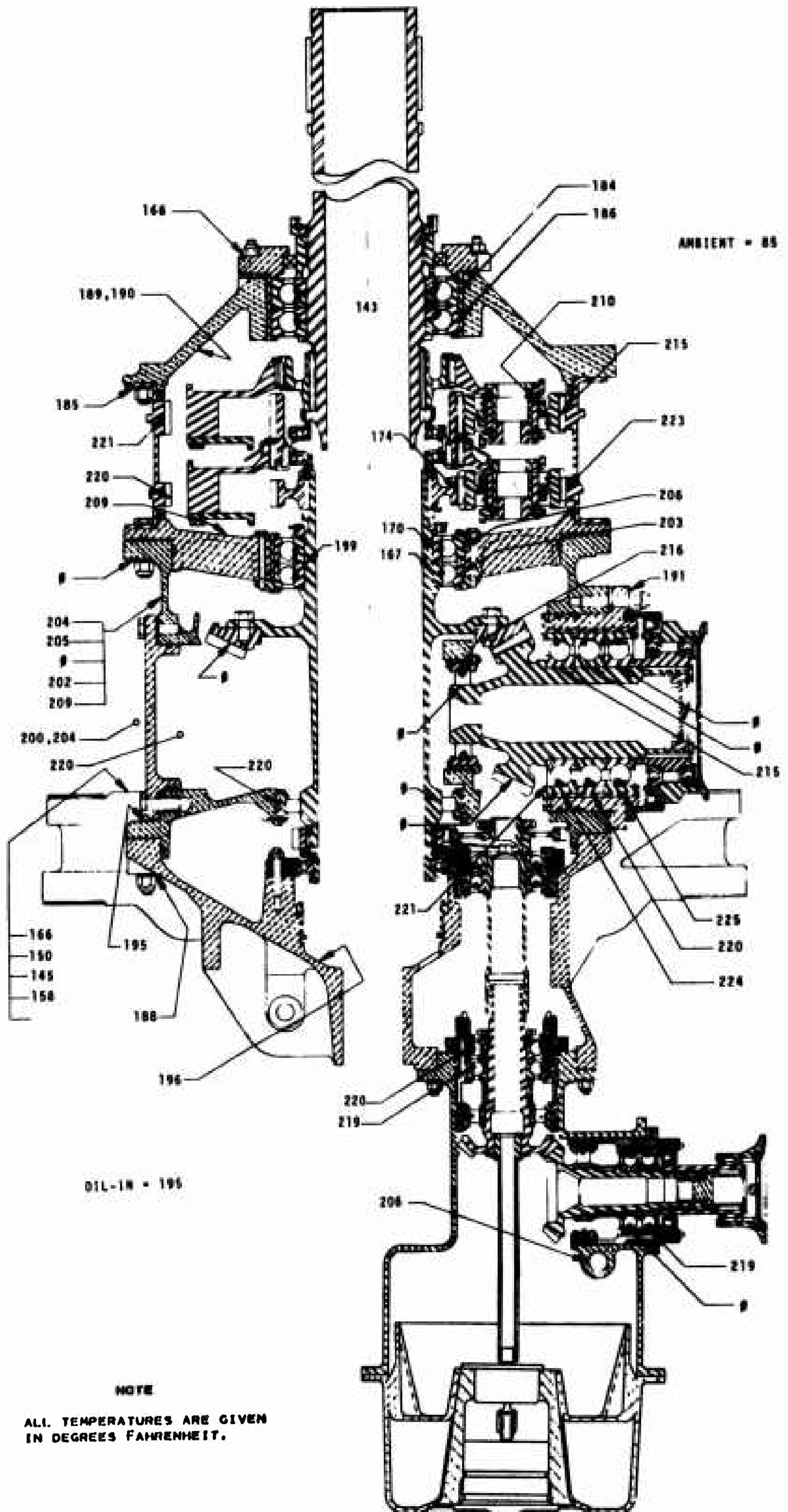

Figure 20. Phase I-2, step 1 transmission temperatures at $200^{\circ}F \pm 5^{\circ}F$ oil-in, 950 horsepower, normal lubrication.

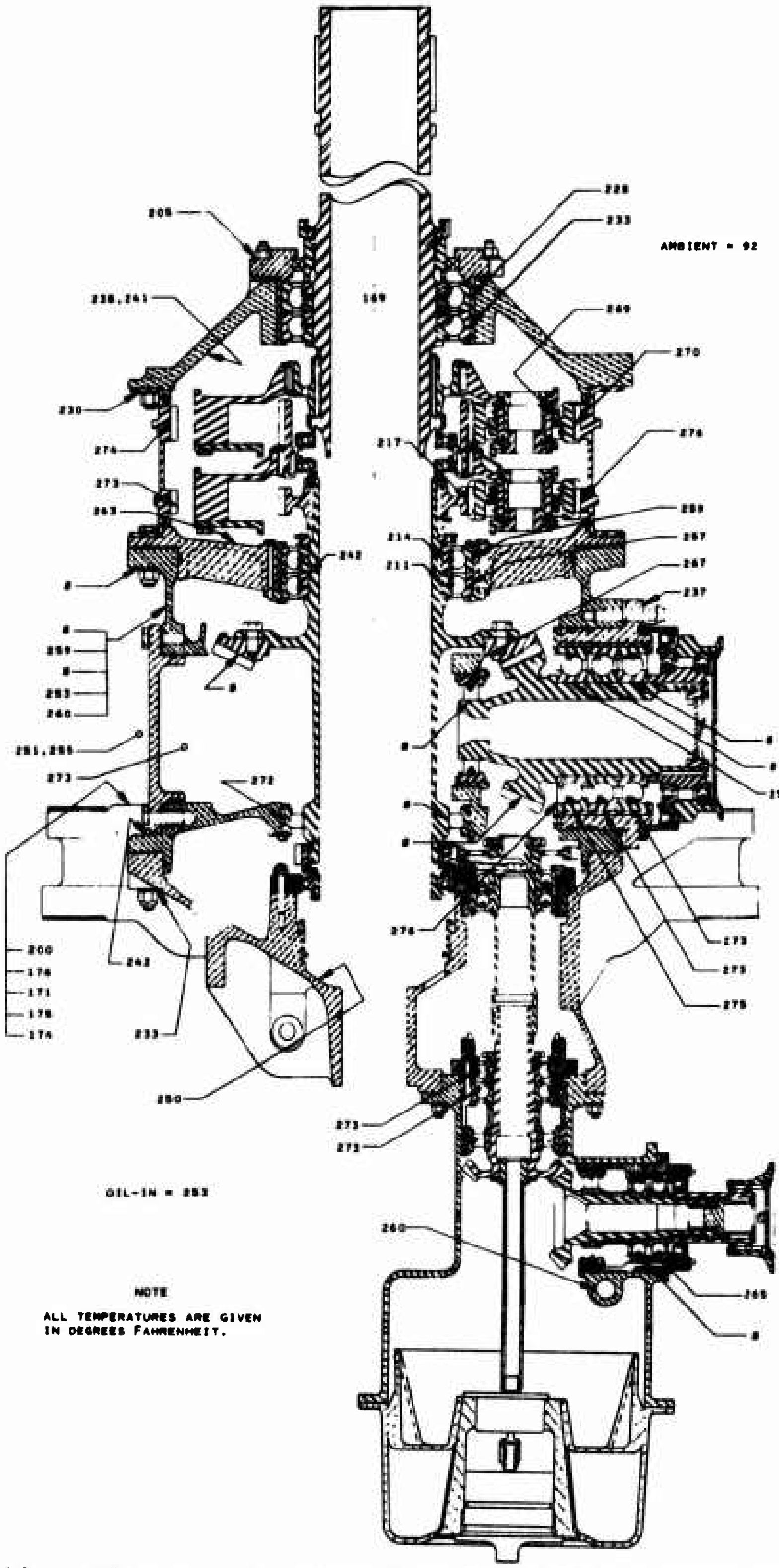

Figure 21. Phase I-2, step 2 transmission temperatures at $250^{\circ}F \pm 5^{\circ}F$ oil-in, 950 horsepower, normal lubrication.

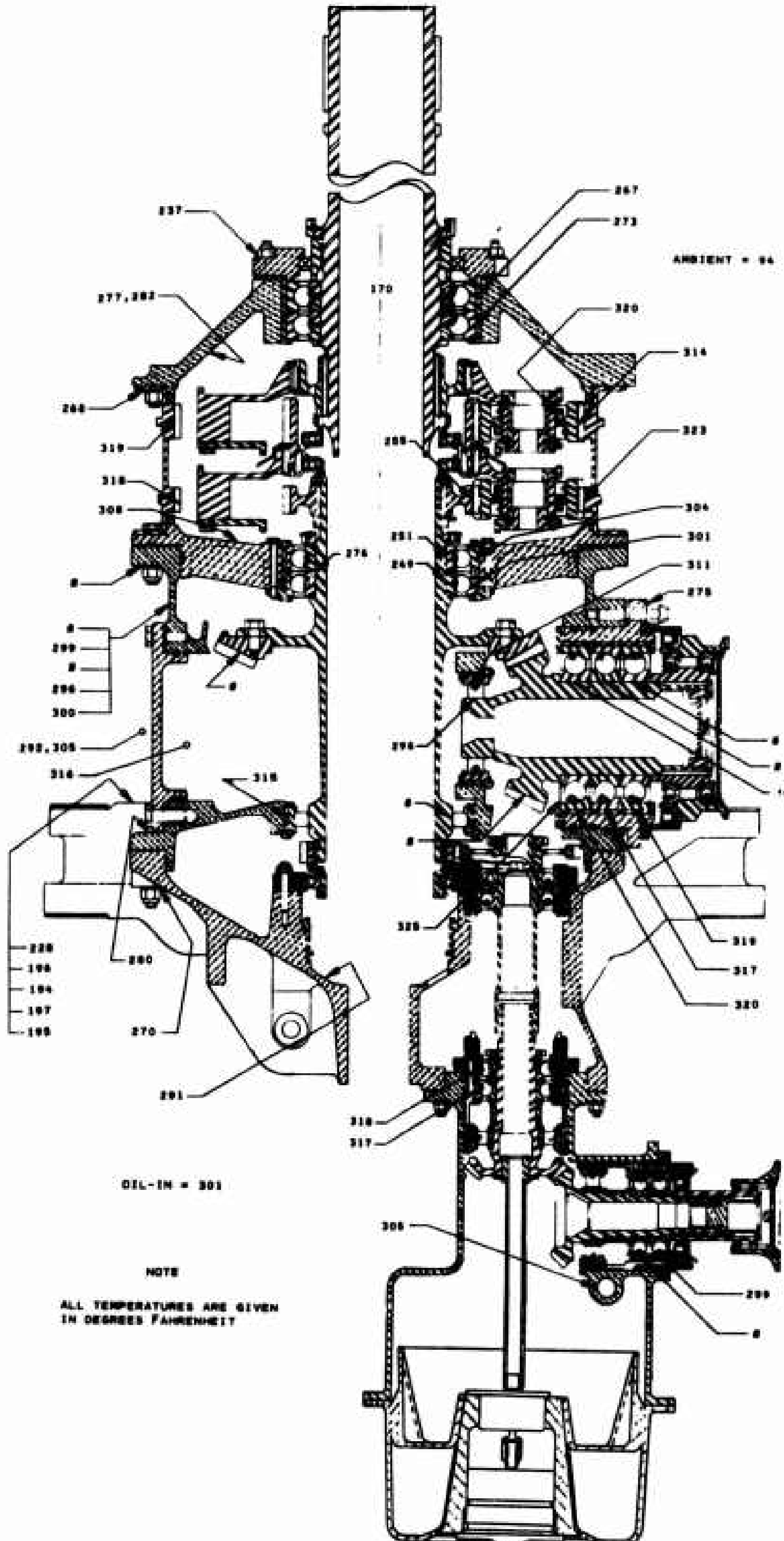

Figure 22. Phase I-2, step 3 transmission temperatures at $300^{\circ}F \pm 5^{\circ}F$ oil-in, 950 horsepower, normal lubrication.

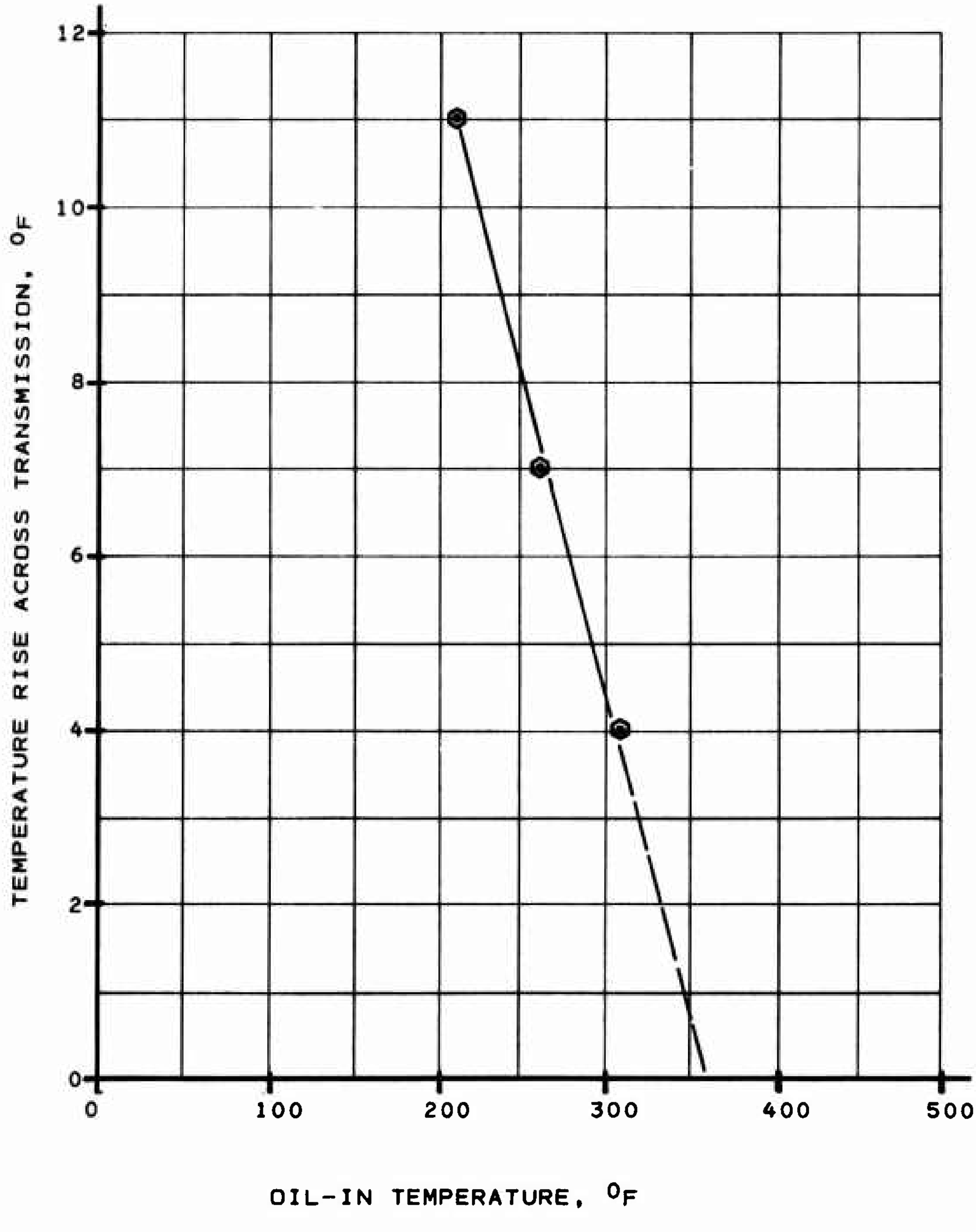

Figure 23. Oil cooler requirements of HST with stubbed mast at 950 horsepower.

- An oil deflector was bonded around the inside perimeter of the upper sump case to direct all of the oil returning from the transmission into the emergency sump.
- The check valve that had been installed backwards in Phase I testing was reinstalled properly.

Figure 24 and Table 3 define the thermocouple locations for Phase II and Phase III testing.

The transmission was reinstalled in the test stand. Temperatures were recorded at intervals of 2 minutes or less. Pressure, oil flow, main rotor mast torque, tail rotor mast torque, and input rpm were recorded at intervals of 6 minutes or less.

5.3.2 Phase II Test Objectives

The transmission was test run per Phase I-3 of Table 4 and as described in Table 5 to meet the following specific objectives:

- To determine the rate at which the oil was lost past the emergency sump.
- To compare the amount of heat generated in the transmission during normal lubrication to that generated during emergency lubrication.
- To obtain a thermal baseline on the outboard bearing of the input triplex and to obtain thermal data on the transmission with the standard mast installed and lift and shear loads applied for comparison with the thermal data obtained during Phase I tests with the stubbed mast and no shear and lift loads applied.
- To determine the length of time the oil cooler can be fully bypassed before any monitored component reaches 350°F.
- To investigate the transmission response to the emergency lubrication system at 950 input horsepower prior to the 60-minute emergency lubrication run.
- To investigate the transmission response to the emergency lubrication system at 1134 input horsepower.

5.3.3 Results of Phase II Tests

The transmission was successfully run per Phase I-3 of Table 4. With the transmission operating at 6600 rpm and minimum input horsepower under normal lubrication with the oil cooler bypassed, the oil-in temperature stabilized at 246°F. The maximum recorded temperature at this stabilized condition was 266°F which was the temperature of the oil out of the input triplex

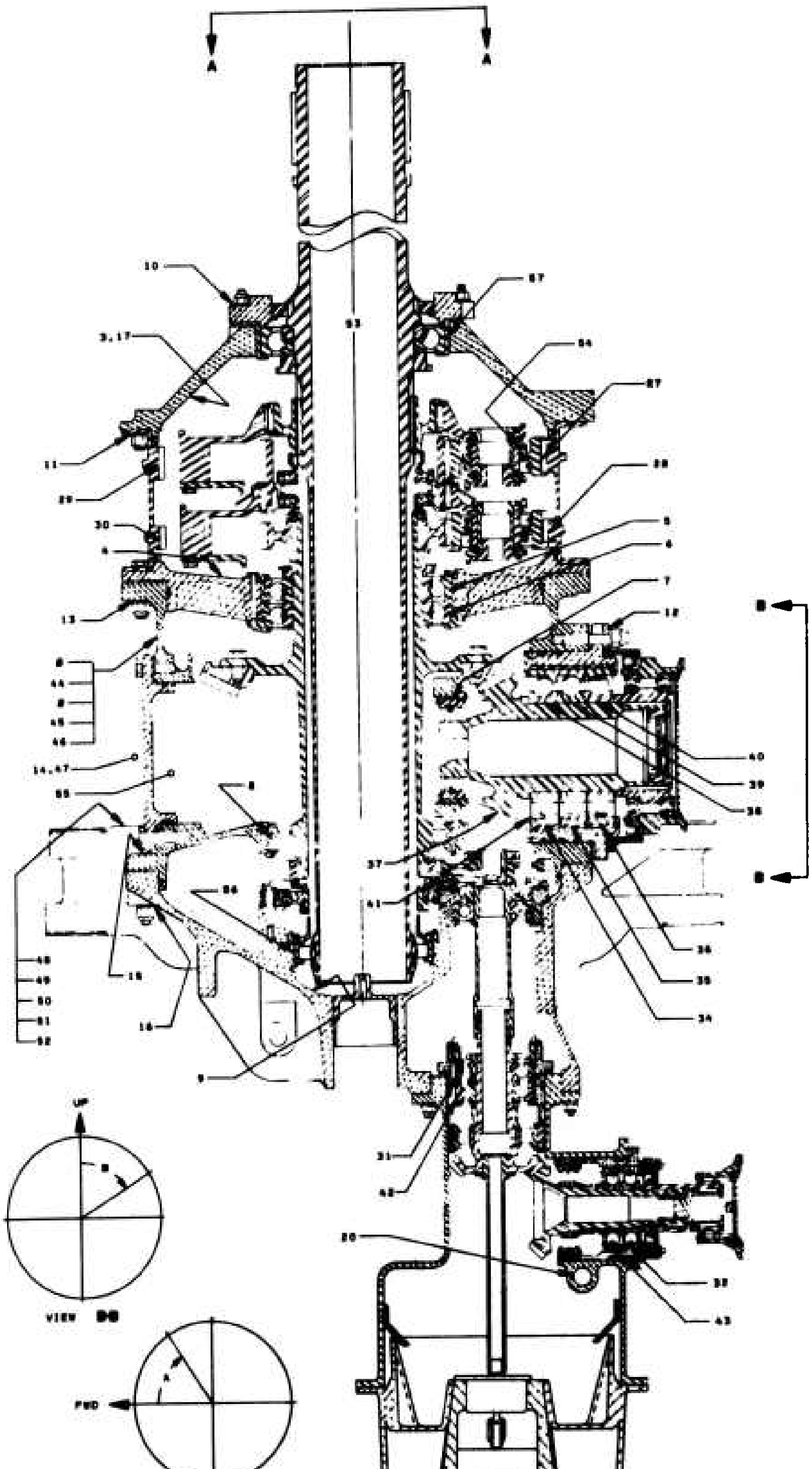

Figure 24. Thermocouple locations for Phase II and Phase III testing.

TABLE 4. BUILDUP VERIFICATION AND THERMAL BASELINE TEST LOAD AND SPEED SCHEDULE FOR TEST I

Phase no.[1]	Step	Time (hrs)	Normal lube with oil cooler	Normal lube no. oil cooler	Emergency lube	Transmission input (rpm)	Transmission input (hp[2])	Main rotor (hp)	Tail rotor (hp)	Transmission oil-in temp[3] (°F)
I-1	1	.1	yes	no	no	4000	minimum	minimum	minimum	180
	2	.1	yes	no	no	4400	130	108	20	180
	3	.1	yes	no	no	4800	141	119	20	180
	4	.1	yes	no	no	5200	153	131	20	180
	5	.1	yes	no	no	6400	440	413	20	180
	6	.1	yes	no	no	7040	827	794	20	180
	7	.1	yes	no	no	6600	775	713	50	180
	8	.1	yes	no	no	6400	840	752	75	180
	9	.1	yes	no	no	6400	990	875	100	180
	10	.1	yes	no	no	6400	1100	1063	20	180
	11	1.0	yes	no	no	6600	1134	1065	51	180
I-2	1	[4]	yes	no	no	6600	950	910	25	200
	2	[4]	yes	no	no	6600	950	910	25	250
	3	[4]	yes	no	no	6600	950	910	25	300
	4	[4]	yes	no	no	6600	950	910	25	350
	5	[4]	yes	no	no	6600	1134	1065	51	350
I-3	1	-	no	yes	no	6600	minimum	minimum	minimum	[5]
	2	-	yes	no	no	6600	minimum	minimum	minimum	230
	3	1.0	no	no	yes	6600	minimum	minimum	minimum	-

[1]No lift or shear loads were applied to the mast since the stubbed mast was being used.

[2]This value includes 1.6% of the transmitted main rotor and tail rotor hp.

[3]These are stabilized oil temperatures (stabilized defined as 1°F or less change in .1 hour).

[4]Run until specified stabilized oil temperature is attained.

[5]Run until the oil temperature stabilizes.

TABLE 5. THERMAL BASELINE AND EMERGENCY RESPONSE TESTING LOAD AND SPEED SCHEDULE

Phase no.[1]	Step	Time (hrs)	Normal lube with oil cooler	Normal lube no. oil cooler	Emergency lube	Transmission input (rpm)	Transmission input (hp[2])	Main rotor (hp)	Tail rotor (hp)	Transmission oil-in temp[3] (°F)
II-1	1	[4]	yes	no	no	6600	950	910	25	200
	2	[4]	yes	no	no	6600	950	910	25	250
	3	[4]	yes	no	no	6600	950	910	25	300
	4	[4]	yes	no	no	6600	950	910	25	350
	5	[4]	yes	no	no	6600	1134	1065	51	350
II-2	1	.3	yes	no	no	6600	950	910	25	230
	2	[4]	no	yes	no	6600	950	910	25	300
II-3	1	.3	yes	no	no	6600	950	910	25	230
	2	.2	no	no	yes	6600	950	910	25	-
	3	.5	yes	no	no	6600	700	674	15	-
II-4	1	.3	yes	no	no	6600	1134	1065	51	230
	2	.2	no	no	yes	6600	1134	1065	51	-
	3	.5	yes	no	no	6600	700	572	15	-

[1]5600 lb lift load and 480 lb shear load applied to the main rotor mast.

[2]This value includes 1.6% of the transmitted main rotor and tail rotor hp.

[3]These are stabilized oil temperatures (stabilized defined as 1°F or less change in .1 hour).

[4]Run until specified stabilized oil temperature is attained.

bearing. Transmission temperatures at this stabilized condition are shown in Figure 25. Figure 26 shows the recorded temperatures after 1 hour of transmission operation at 6600 rpm and minimum input horsepower utilizing just the emergency lubrication system. The oil-in temperature stabilized at 219°F, and all recorded temperatures for this run were cooler than those recorded for the Phase I-3, step 1 run under normal lubrication with the oil cooler bypassed. Under emergency lubrication, there was less oil circulating in the transmission and thus less windage and churning losses. Therefore, at minimum input horsepower, the transmission ran cooler under emergency lubrication than under normal lubrication with no oil cooler. The emergency oil pump maintained a pressure reading of 13 psi throughout the emergency run. Following the test run, the oil was drained from the emergency sump, and it was found to contain 1380 ml (1.46 qt).

Next, the transmission was run per Phase II-1 of Table 5. Figures 27, 28, and 29 show the transmission temperatures at the stabilized oil-in temperatures for steps 1, 2, and 3, respectively, of Phase II-1.

Some problems were encountered in trying to achieve the 350°F stabilized oil-in temperature of step 4. This was due to the fact that as the oil temperature increased, the input oil seal lost its ability to seal adequately, and a significant quantity of oil leaked past the seal. When the input oil temperature reached approximately 315°F, so much oil was being lost past the seal that a quart of oil had to be added to the transmission every 15 to 20 minutes. Adding a quart of oil would drop the oil-in temperature back down to around 300°F. By the time the oil heated up again to the 315°F range, another quart of oil would be required by the transmission. Finally, since the higher temperatures could not be attained as long as new oil was added every 15 to 20 minutes, the decision was made to run without adding oil until the oil-in temperature stabilized or until the oil pressure dropped below 40 psi, whichever occurred first (normal oil pressure is 55 psi). Under these conditions, the oil-in temperature finally stabilized at 321°F with the oil pressure reading 44 psi. Transmission temperatures at this stabilized condition are shown in Figure 30. The transmission oil level was approximately 3 quarts low at this time. Thus, utilization of these data must be tempered with the fact that both oil pressure and oil level were below normal operating conditions.

Since the 350°F stabilized oil-in temperature of step 4 could not be attained, it was decided to lower the required stabilized oil-in temperature of step 5 to 300°F rather than attempt to reach 350°F at 1134 input horsepower. Figure 31 shows the transmission temperatures at the 300°F.

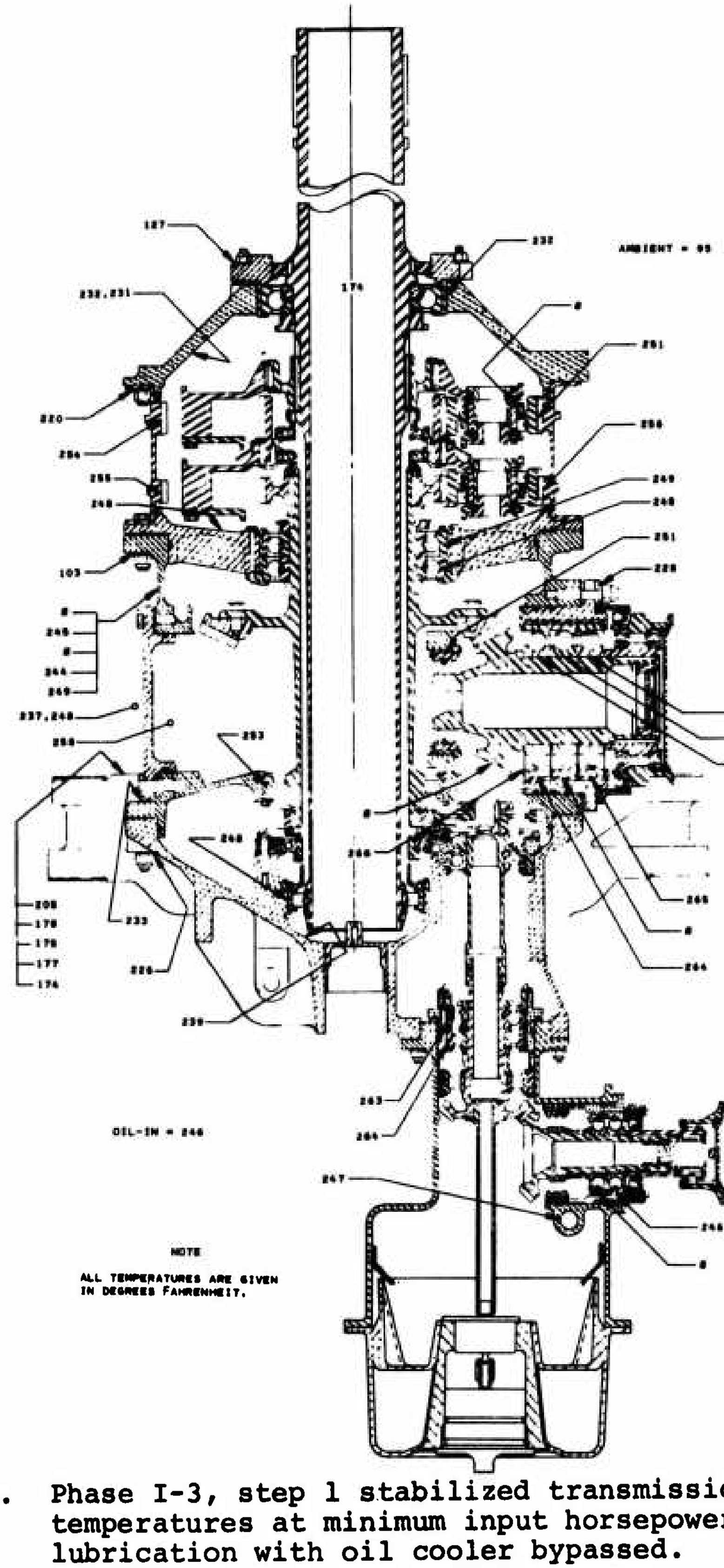

Figure 25. Phase I-3, step 1 stabilized transmission temperatures at minimum input horsepower, normal lubrication with oil cooler bypassed.

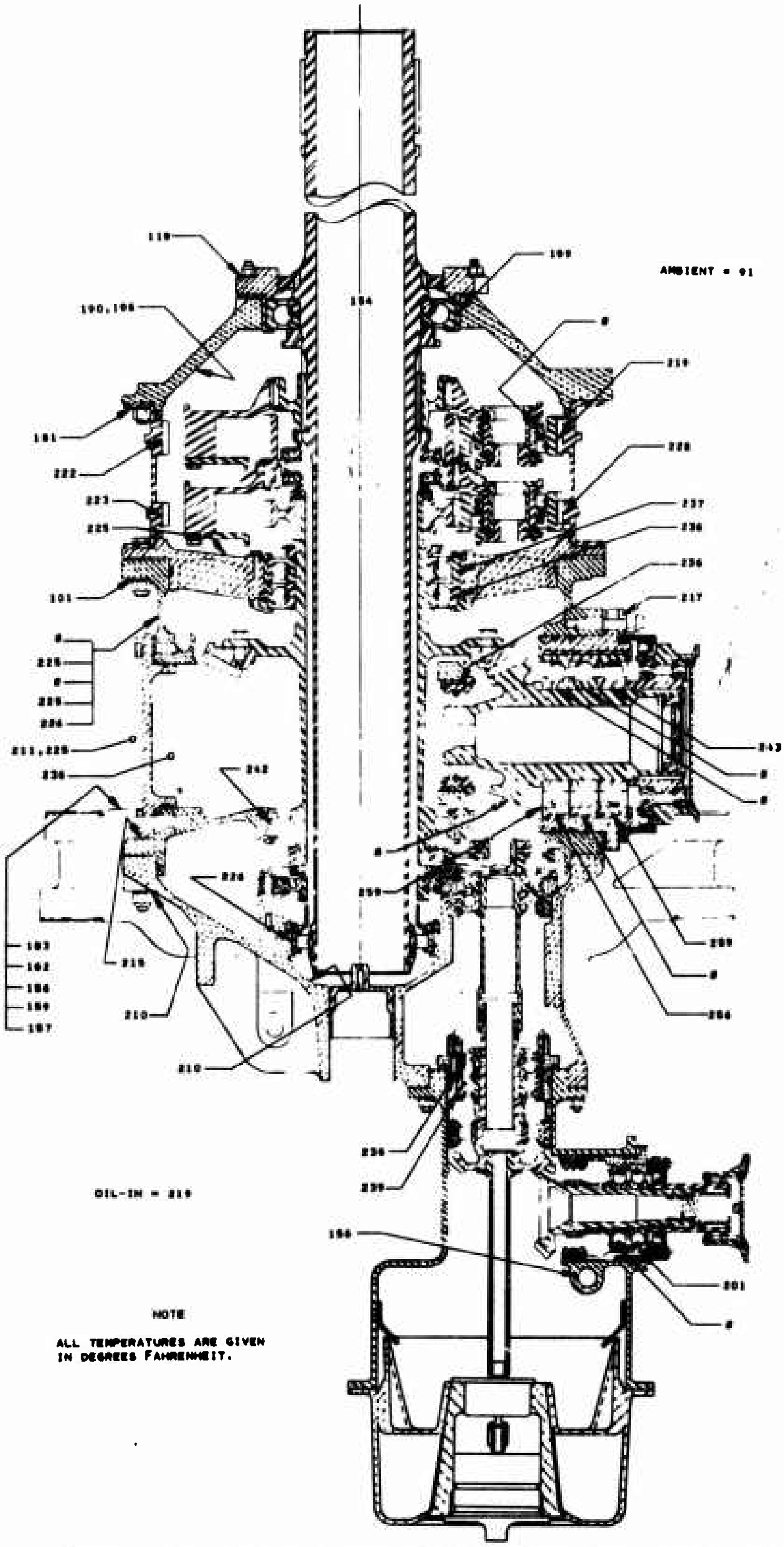

Figure 26. Phase I-3, step 3 transmission temperatures after one hour on emergency lubrication at minimum input horsepower.

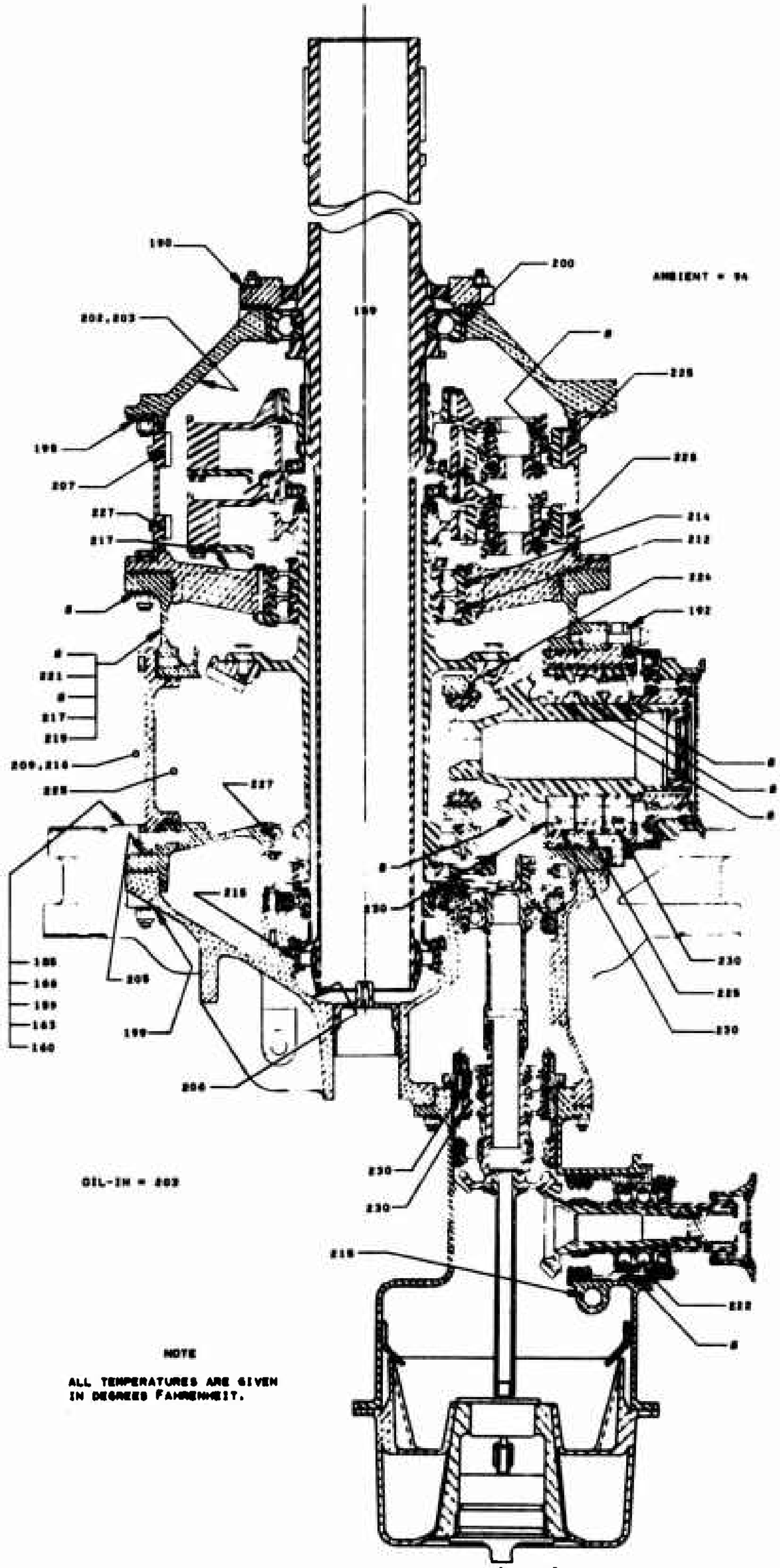

Figure 27. Phase II-1, step 1 transmission temperatures at $200^{o}F \pm 5^{o}F$ oil-in, 950 horsepower, normal lubrication.

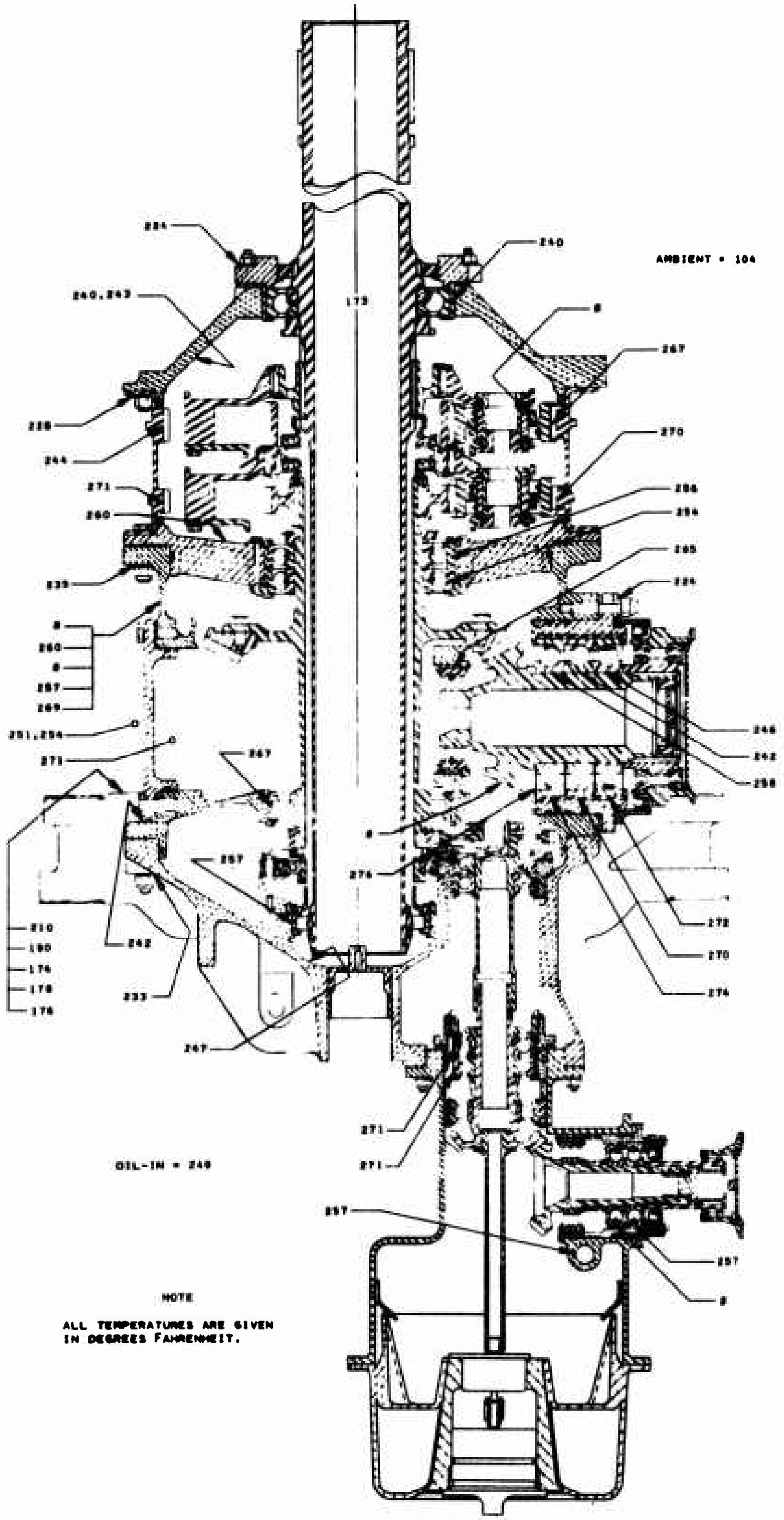

Figure 28. Phase II-1, step 2 transmission temperatures at 250°F ± 5°F oil-in, 950 horsepower, normal lubrication.

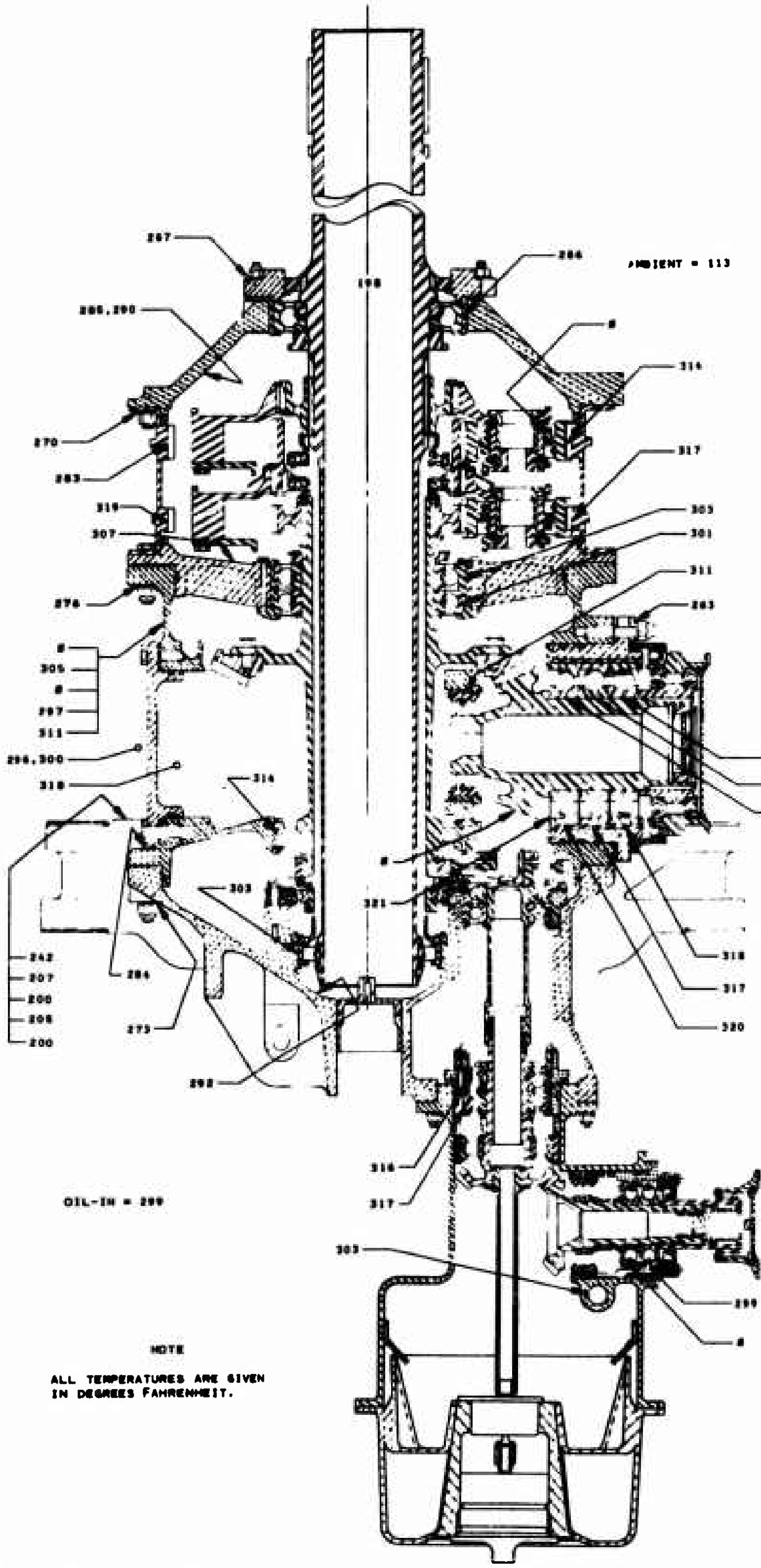

Figure 29. Phase II-1, step 3 transmission temperatures at $300^{\circ}F \pm 5^{\circ}F$ oil-in, 950 horsepower, normal lubrication.

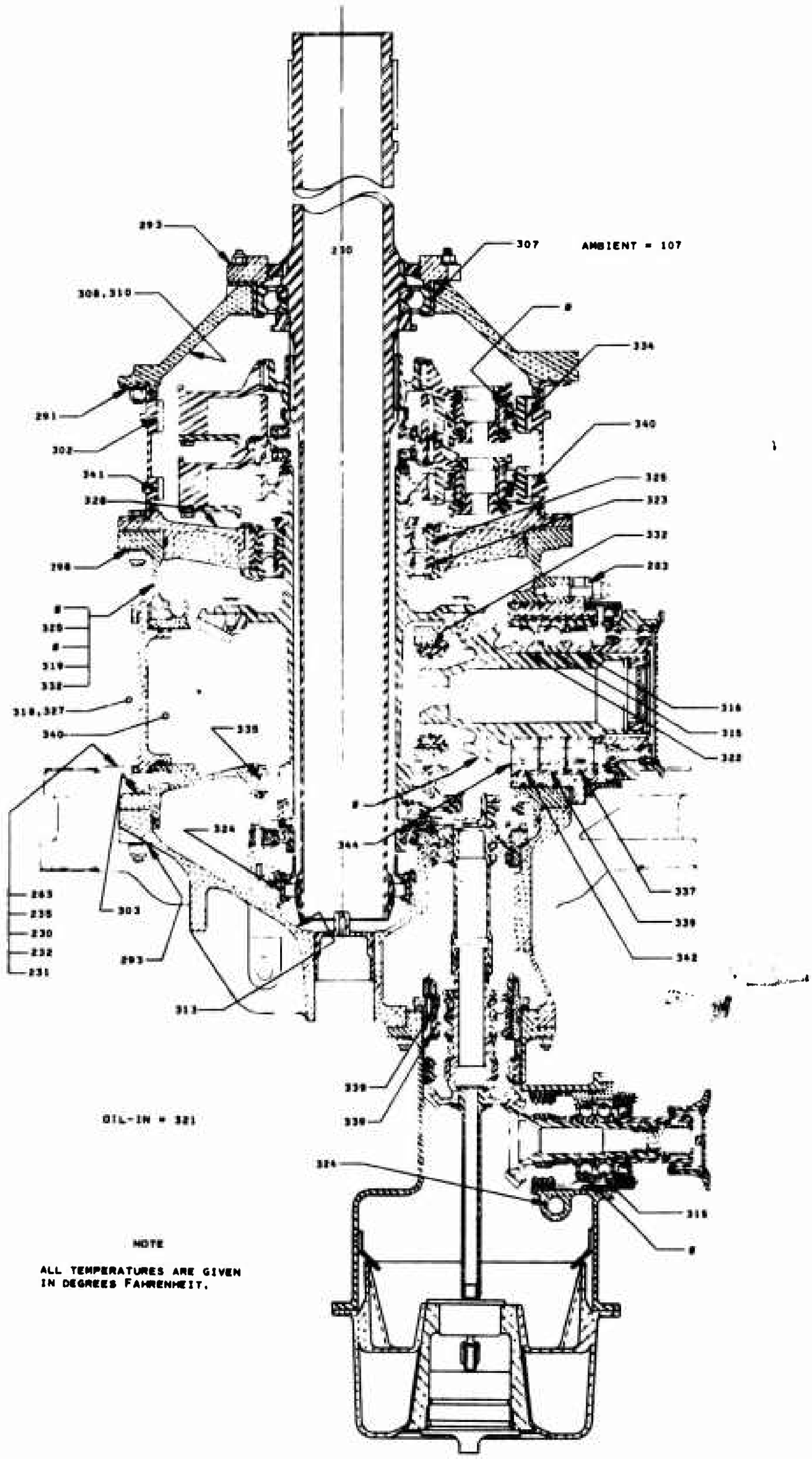

Figure 30. Phase II-1, step 4 transmission temperatures at 950 horsepower, normal lubrication with oil cooler bypassed.

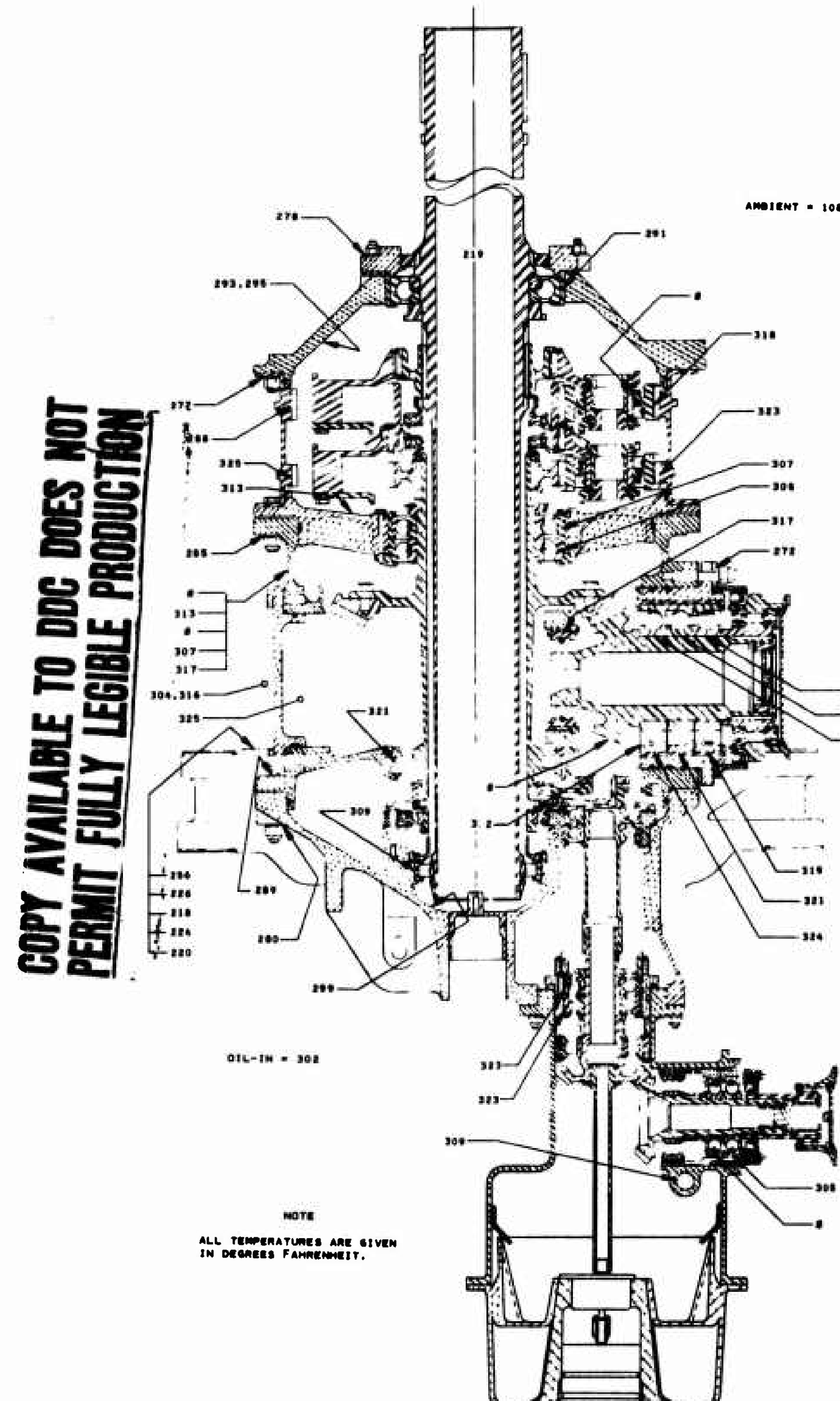

Figure 31. Phase II-1, step 5 transmission temperatures at $300^{\circ}F \pm 5^{\circ}F$ oil-in, 1134 horsepower, normal lubrication.

Phase II-2 was amended due to the difficulty of attaining the 350°F temperature. Rather than run until the hottest monitored component reached 350°F, it was decided to run until the oil-in temperature reached 300°F. A plot of oil-in temperature versus time is shown in Figure 32.

Phase II-3 was an emergency response run at 950 input horsepower and 6600 rpm with 25 horsepower through the tail rotor. The transmission was run on normal lubrication until the oil-in temperature stabilized at 230°F ±5°F. A ballistic hit resulting in total loss of normal lubrication was then simulated, and the transmission was allowed to run for 12 minutes on emergency lubrication. This test was performed 3 times and the results were very consistent. Typical transmission temperatures recorded at the stabilized oil-in temperature of 230°F±5°F just prior to draining the main oil supply are shown in Figure 33. Figure 34 shows typical transmission temperatures after 12 minutes of operation on emergency lubrication. A typical plot of the oil-in temperature versus time for the test runs of Phase II-3 is shown in Figure 35.

Phase II-4 was an emergency response run at 1134 input horsepower and 6600 rpm with 51 horsepower through the tail rotor. The transmission was run on normal lubrication until the oil-in temperature stabilized at 230°F±5°F. A ballistic hit resulting in total loss of normal lubrication was then simulated, and the transmission was allowed to run for 12 minutes on emergency lubrication. Transmission temperatures recorded at the stabilized oil-in temperature of 230°F±5°F just prior to draining the main oil supply are shown in Figure 36. Figure 37 shows the recorded transmission temperatures after 12 minutes of operation on emergency lubrication. A plot of the oil-in temperature versus time of the Phase II-4 test is shown in Figure 35.

5.4 PHASE III TESTS

5.4.1 Phase III Test Configuration

Phase III testing immediately followed Phase II testing, and thus transmission configuration and instrumentation for Phase III were identical to those of Phase II.

Transmission temperatures were recorded at 1-minute intervals throughout Phase III testing. Oil pressure, oil flow rate, main rotor mast torque, tail rotor mast torque, and input rpm were recorded at intervals of 6 minutes or less.

5.4.2 Phase III Test Objectives

The transmission was tested per Table 6 to meet the following specific objectives:

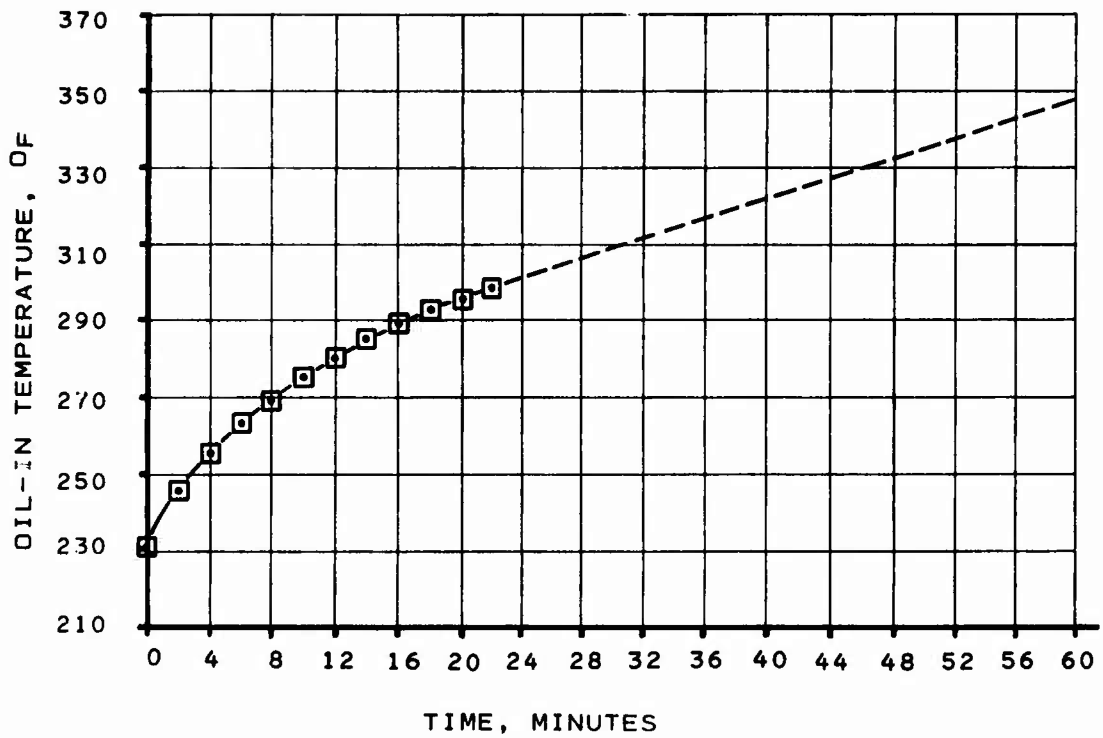

Figure 32. Phase II-2, step 2 oil-in temperature versus time, normal lubrication with oil cooler bypassed at 950 horsepower.

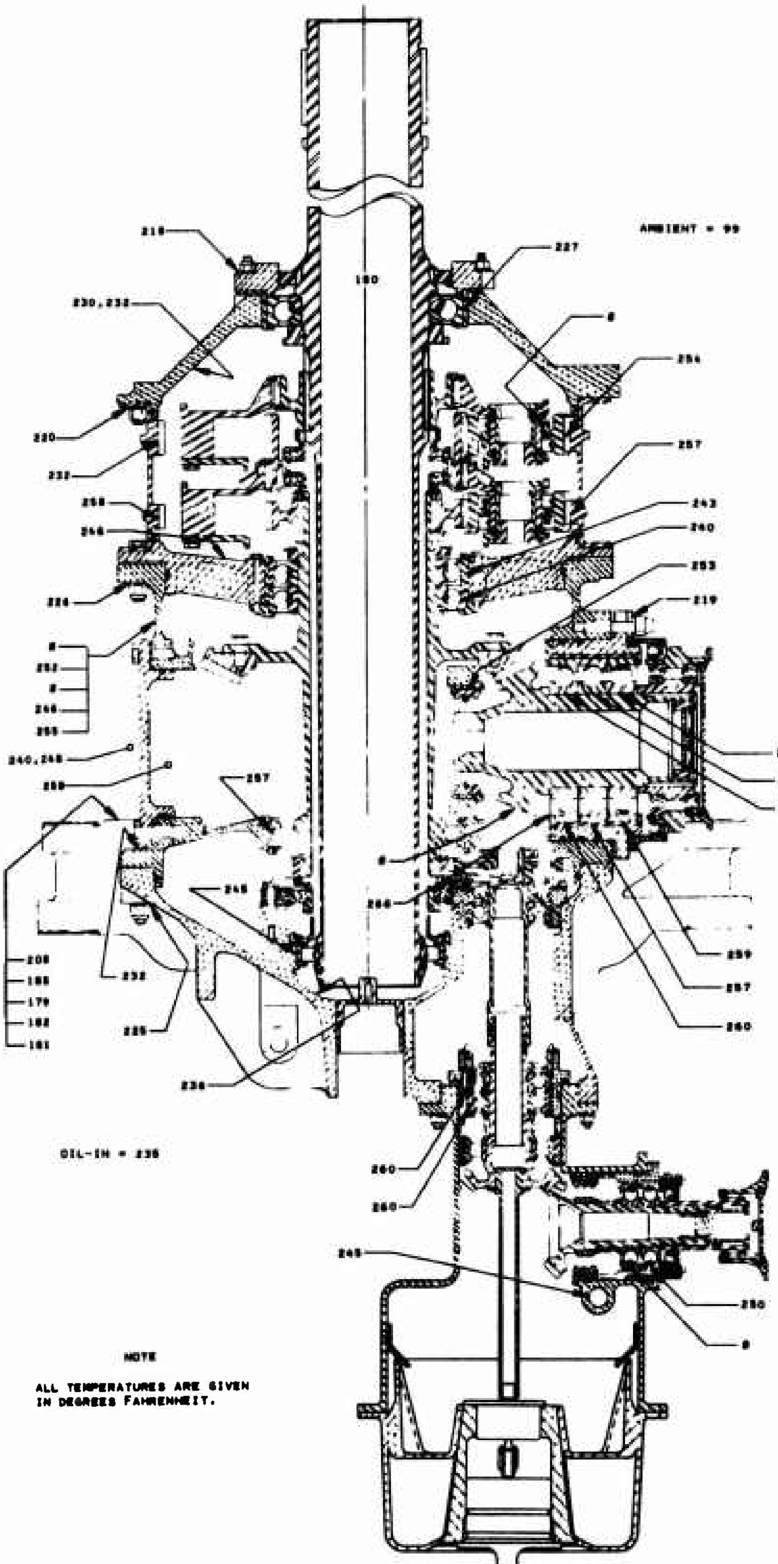

Figure 33. Phase II-3, step 1 typical transmission temperatures at 230°F ± 5°F oil-in, 950 horsepower, normal lubrication.

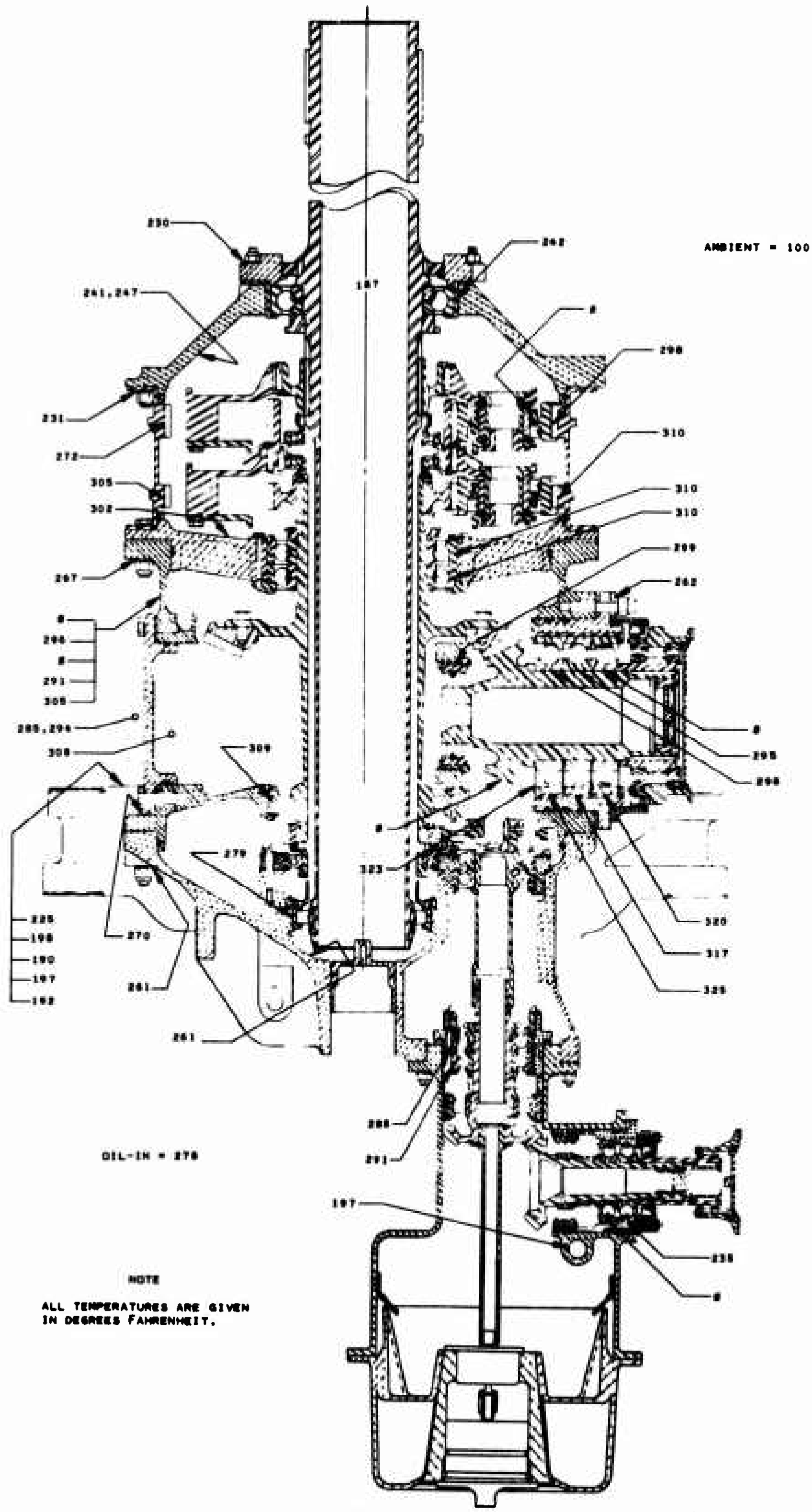

Figure 34. Phase II-3, step 2 typical transmission temperatures after 12 minutes of operation at 950 horsepower on emergency lubrication.

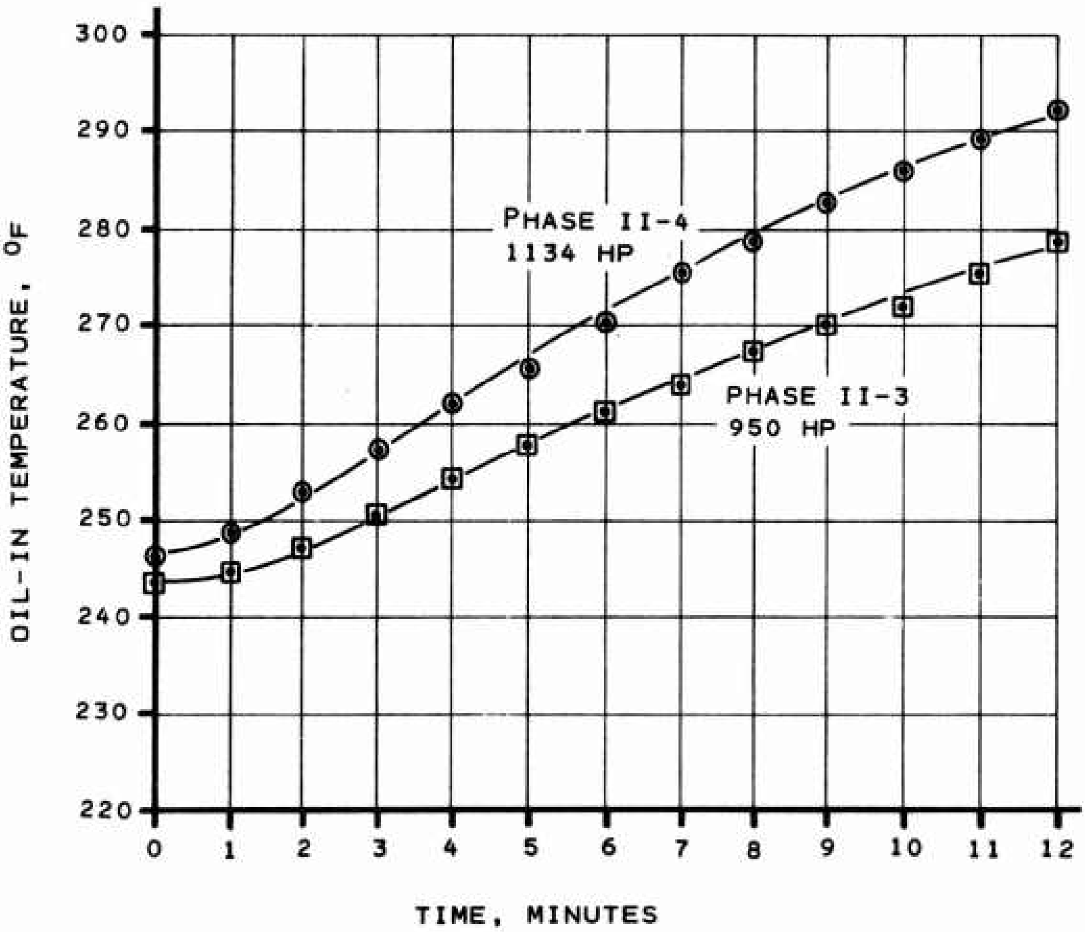

Figure 35. Oil-in temperature versus time during emergency lubrication response runs of Phases II-3 and II-4.

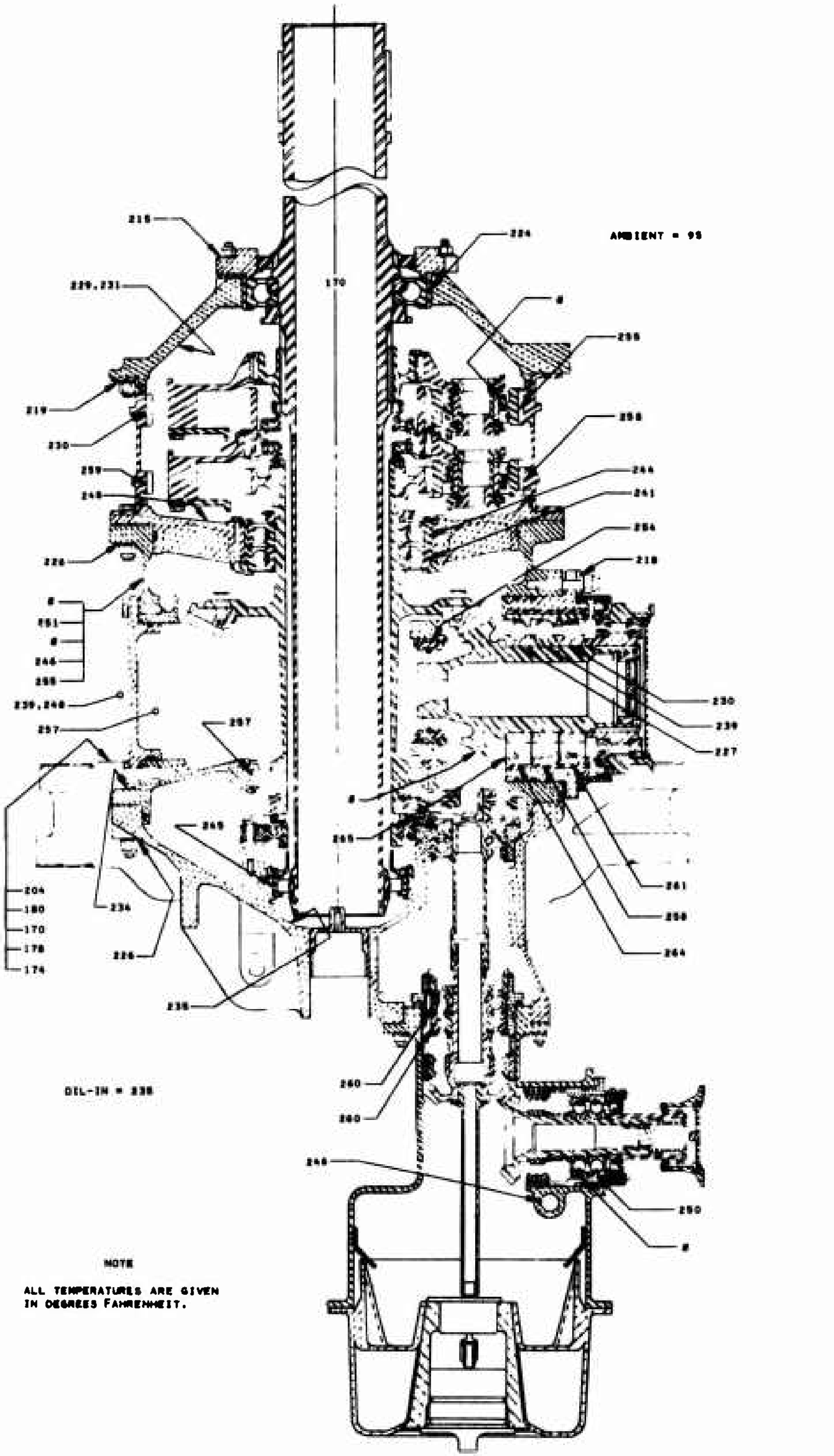

Figure 36. Phase II-4, step 1 transmission temperatures at $230^{\circ}F \pm 5^{\circ}F$ oil-in, 1134 horsepower, just prior to emergency lubrication response run.

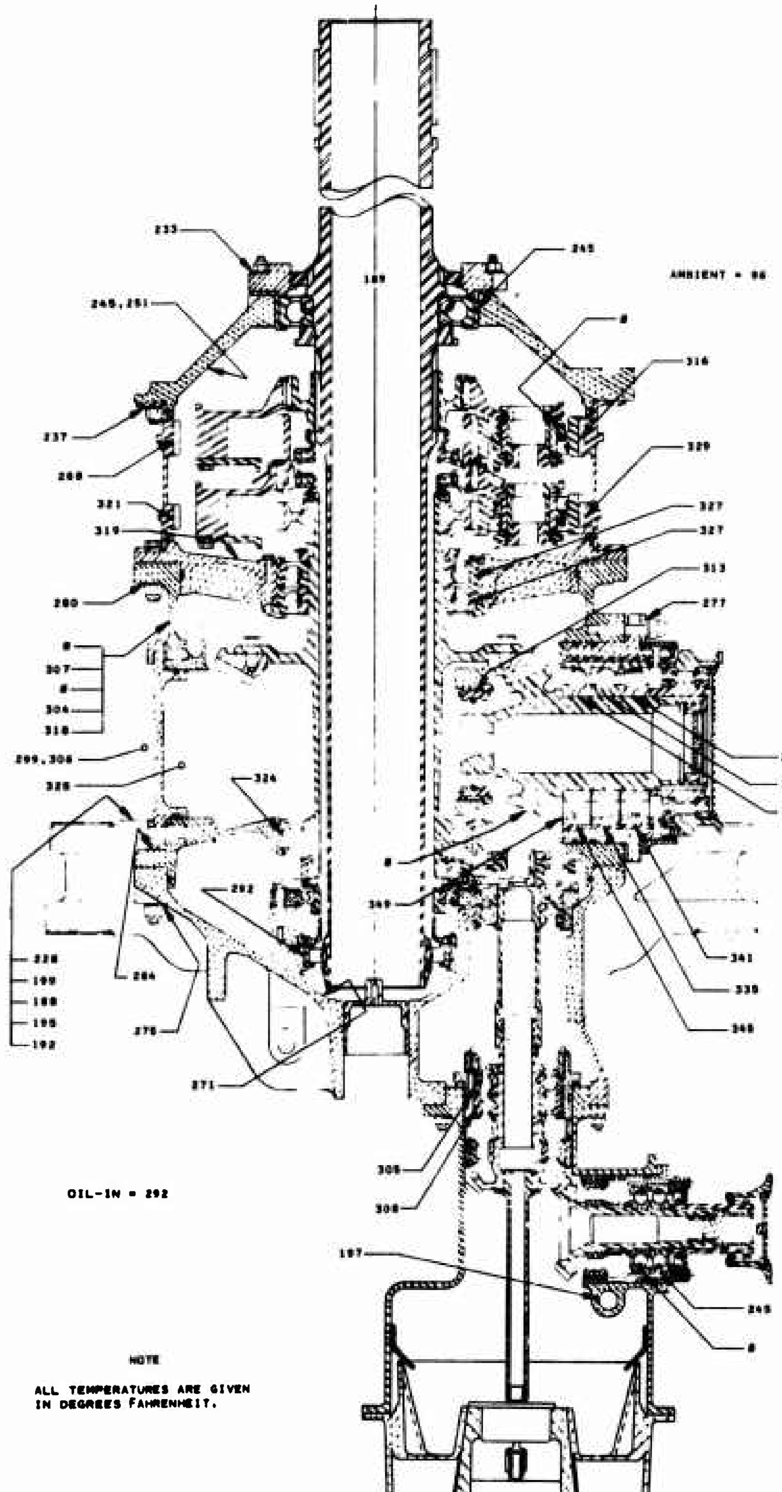

Figure 37. Phase II-4, step 2 transmission temperatures after 12 minutes of operation at 1134 horsepower on emergency lubrication.

TABLE 6. EMERGENCY LUBRICATION 60-MINUTE TESTING LOAD AND SPEED SCHEDULE FOR TEST III

Phase no.[1]	Step	Time (hrs)	Normal lube with oil cooler	Normal lube no. oil cooler	Emergency lube	Transmission input (rpm)	Transmission input (hp[2])	Main rotor (hp)	Tail rotor (hp)	Transmission oil-in temp[3] (°F)
III-1	1	.3	yes	no	no	6600	950	910	25	230
	2	[4]	no	no	yes	6600	950	910	25	-

[1]5600 lb of lift load and 480 lb of shear load applied to the main rotor mast.

[2]This value includes 1.6% of the transmitted main rotor and tail rotor hp.

[3]These are stabilized oil temperatures (stabilized defined as 1°F or less change in .1 hour).

[4]Run until failure occurs.

1. To determine the length of time to failure that the transmission could be operated at 950 input horsepower with the emergency lubrication functioning.

2. To determine the length of time to failure that the transmission could be operated at 950 input horsepower after the emergency lubrication pump stopped pumping, which was defined as the point where the emergency oil pressure drops to zero psi.

A ballistic hit resulting in total loss of lubricant for Phase III-1, step 2 was simulated by closing valves A and B and opening valve C (valves are shown in Figure 19). To further ensure that no oil from the main system could return to the transmission, the oil lines at valves A and B were completely disconnected.

5.4.3 Results of Phase III Tests

The HST ran exactly 4 hours after loss of the normal lubrication system. Figures 38 and 39 show the transmission still in the test stand following the run. Figure 40 shows a plot of the hottest monitored component versus time and a plot of oil-in temperature versus time for the run. At failure, the hottest monitored component was the lower ring gear at 935°F. Figure 40 shows that the emergency oil pressure dropped to zero after 1.4 hours of testing, and thus 2.6 hours of the 4-hour emergency run was without the benefit of circulating oil. Figure 41 shows the transmission temperatures at a stabilized oil-in temperature of 230°F$\pm$5°F just prior to the emergency lubrication run. Figures 42 through 47 show transmission temperatures at selected times during the 4-hour loss of lubrication run. For 17 locations within the transmission plots of the recorded temperature versus time during the loss of lubrication run are shown in Figures 48 through 64. It is significant to note that throughout the test, the outer races of the input triplex bearing remained hotter than the inner races. Failure occurred in the form of complete loss of mast torque due to the stripping of the gear teeth from the lower sun gear.

5.4.4 Summary of Results of Visual Post-Run Inspection

All of the bearings were still functional. The upper mast duplex bearing, the lower mast roller bearing, and the tail rotor drive input and output bearings were still oily when removed from the transmission. All other bearings were dry when removed. All of the M-50 bearings appeared to be in excellent condition. All of the silver-plated steel retainers appeared to be in good condition. The high-temperature operation apparently did not affect the plating. There was no indication of loss of backlash in the upper planetary gears or in the input bevel set.

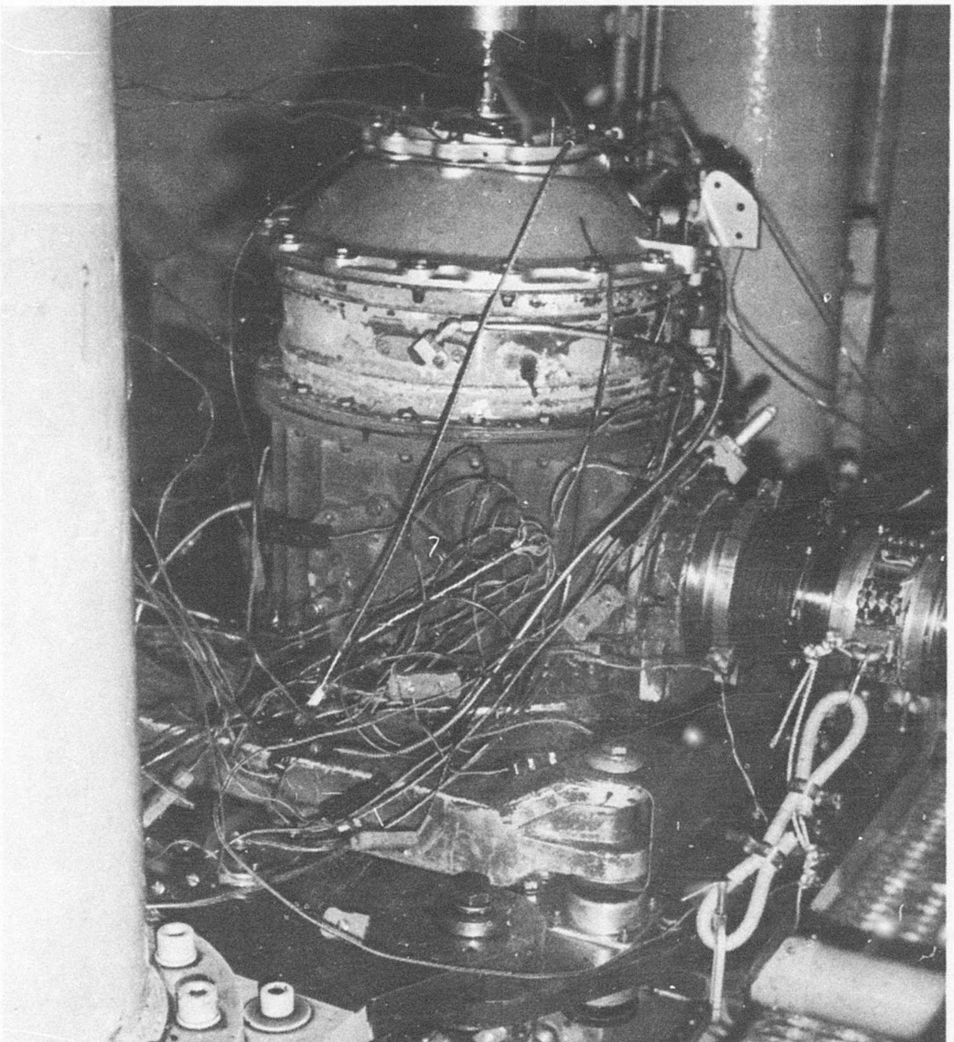

Figure 38. HST in test stand following 4.0-hour emergency lubrication run, left side.

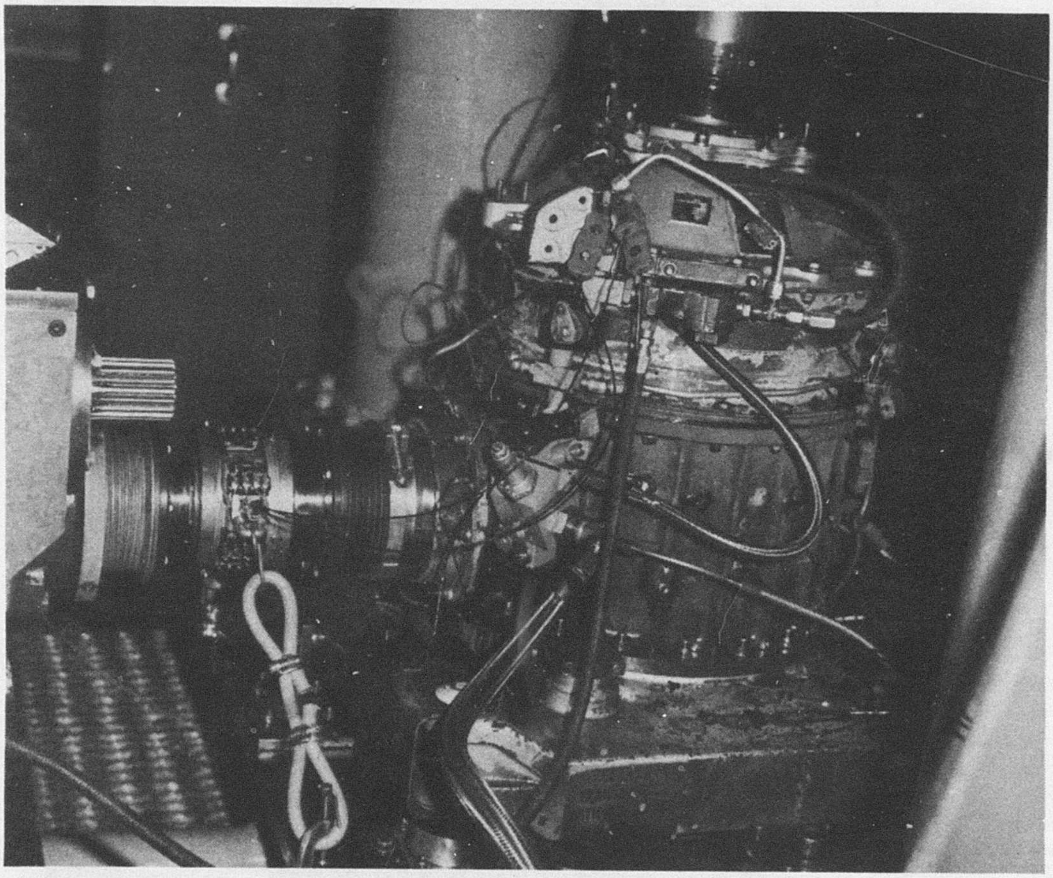

Figure 39. HST in test stand following 4.0-hour emergency lubrication run, right side.

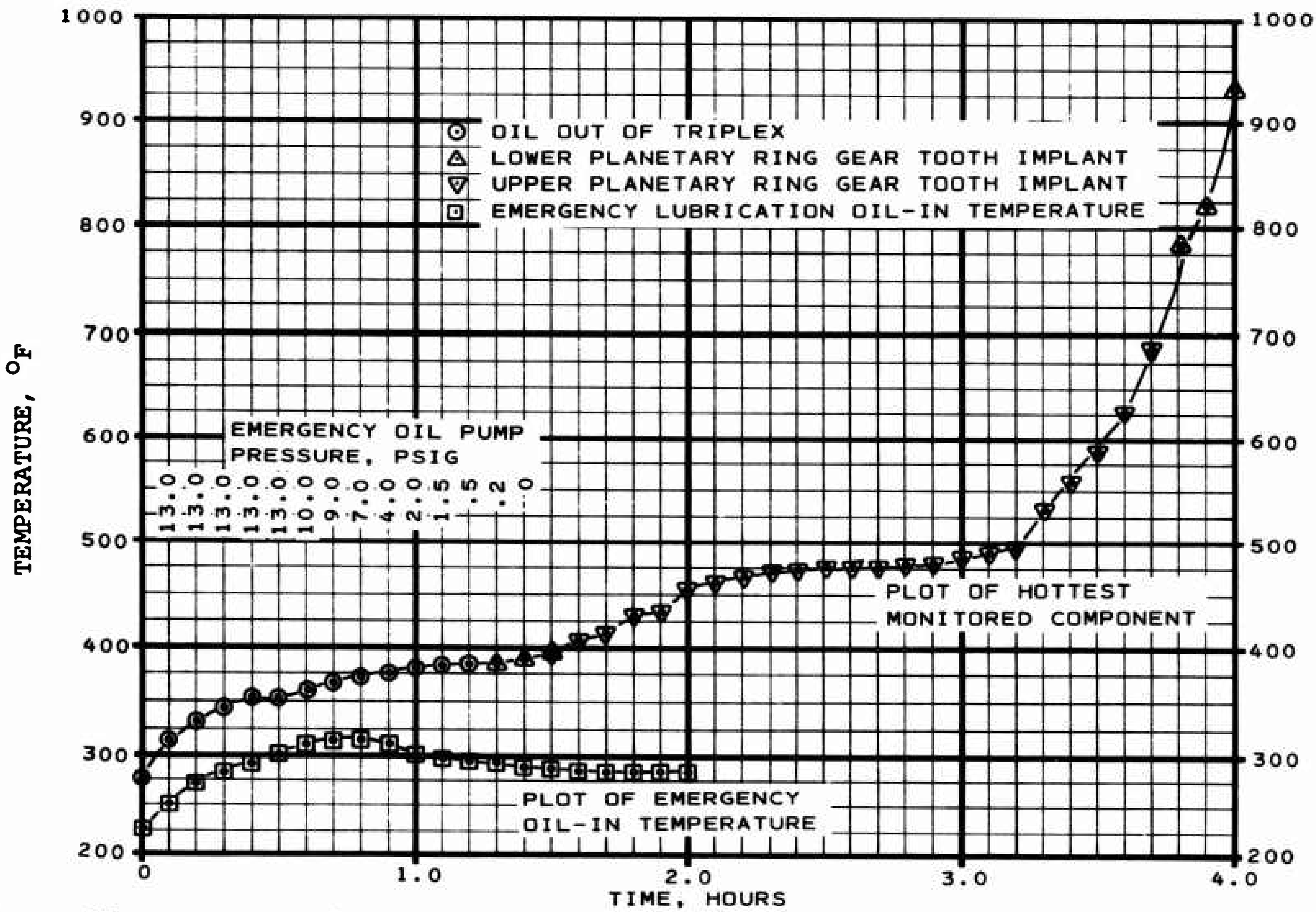

Figure 40. Phase III-1 HST emergency lubrication test results. Test performed at 950 hp input, 6600 rpm, 25 hp to tail rotor, 5600 lb lift and 480 lb shear on mast.

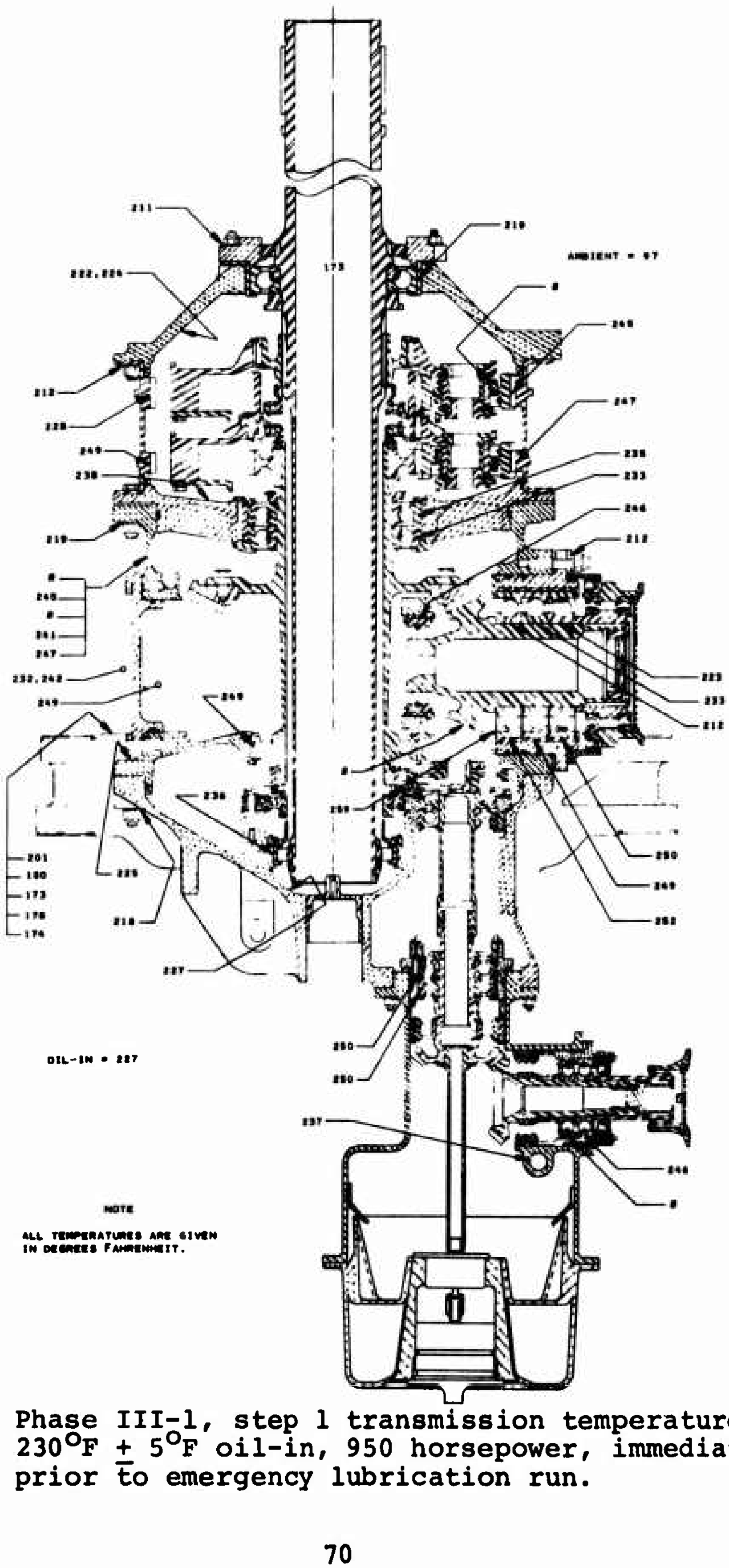

Figure 41. Phase III-1, step 1 transmission temperatures at 230°F $\pm$ 5°F oil-in, 950 horsepower, immediately prior to emergency lubrication run.

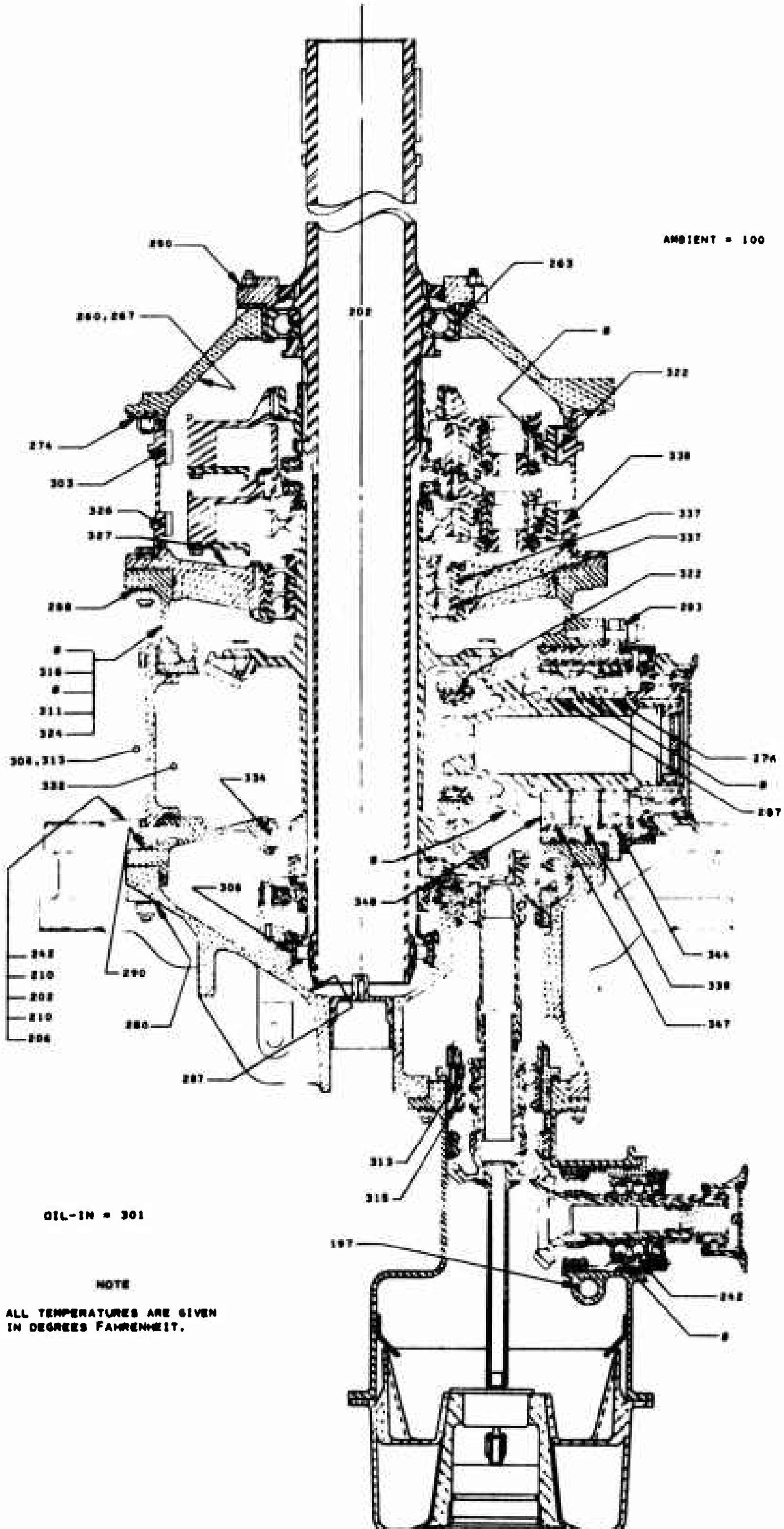

Figure 42. Phase III-1, step 2 transmission temperatures after 30 minutes of emergency operation at 950 horsepower.

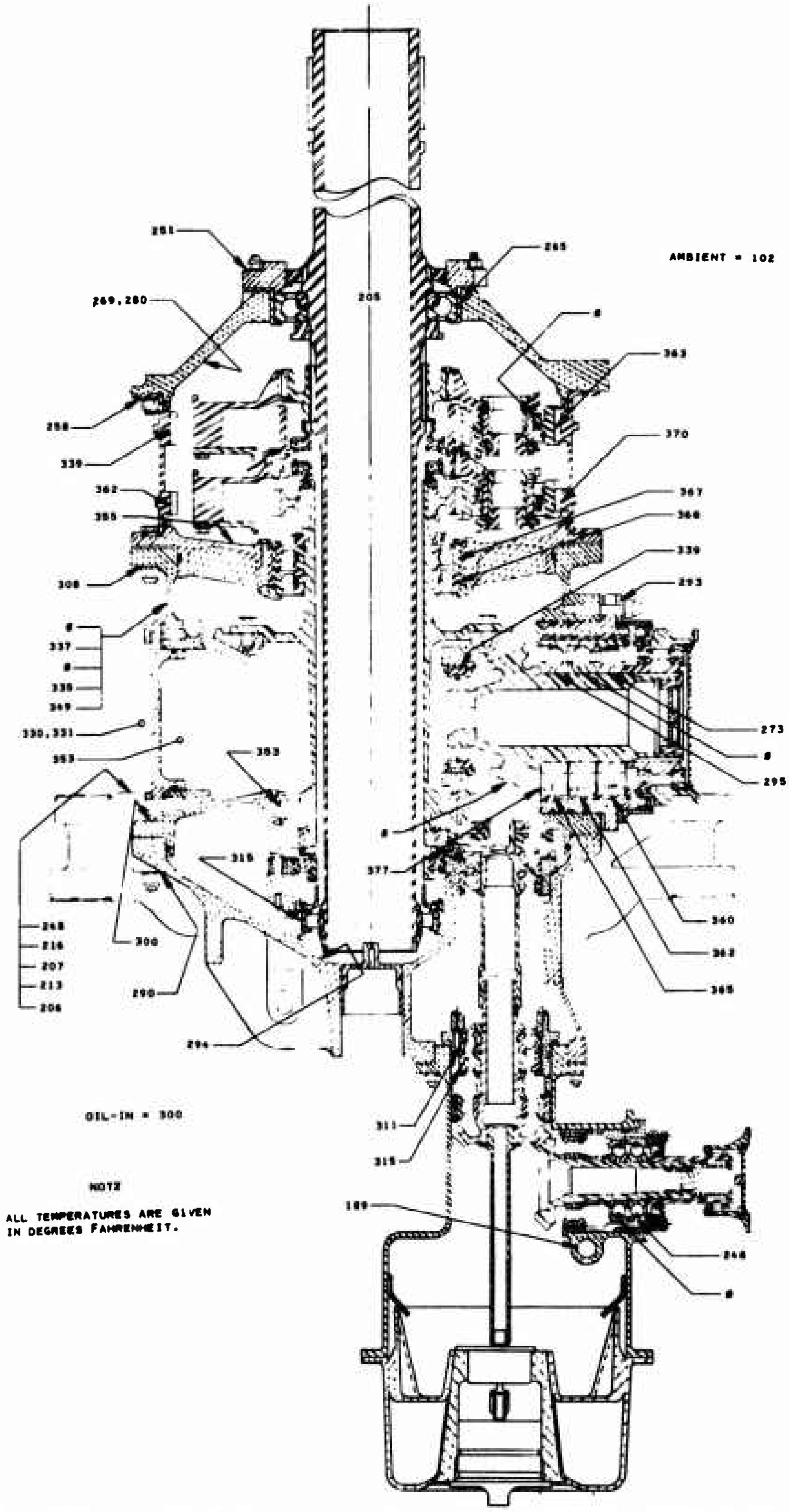

Figure 43. Phase III-1, step 2 transmission temperatures after 1.0 hour of emergency operation at 950 horsepower.

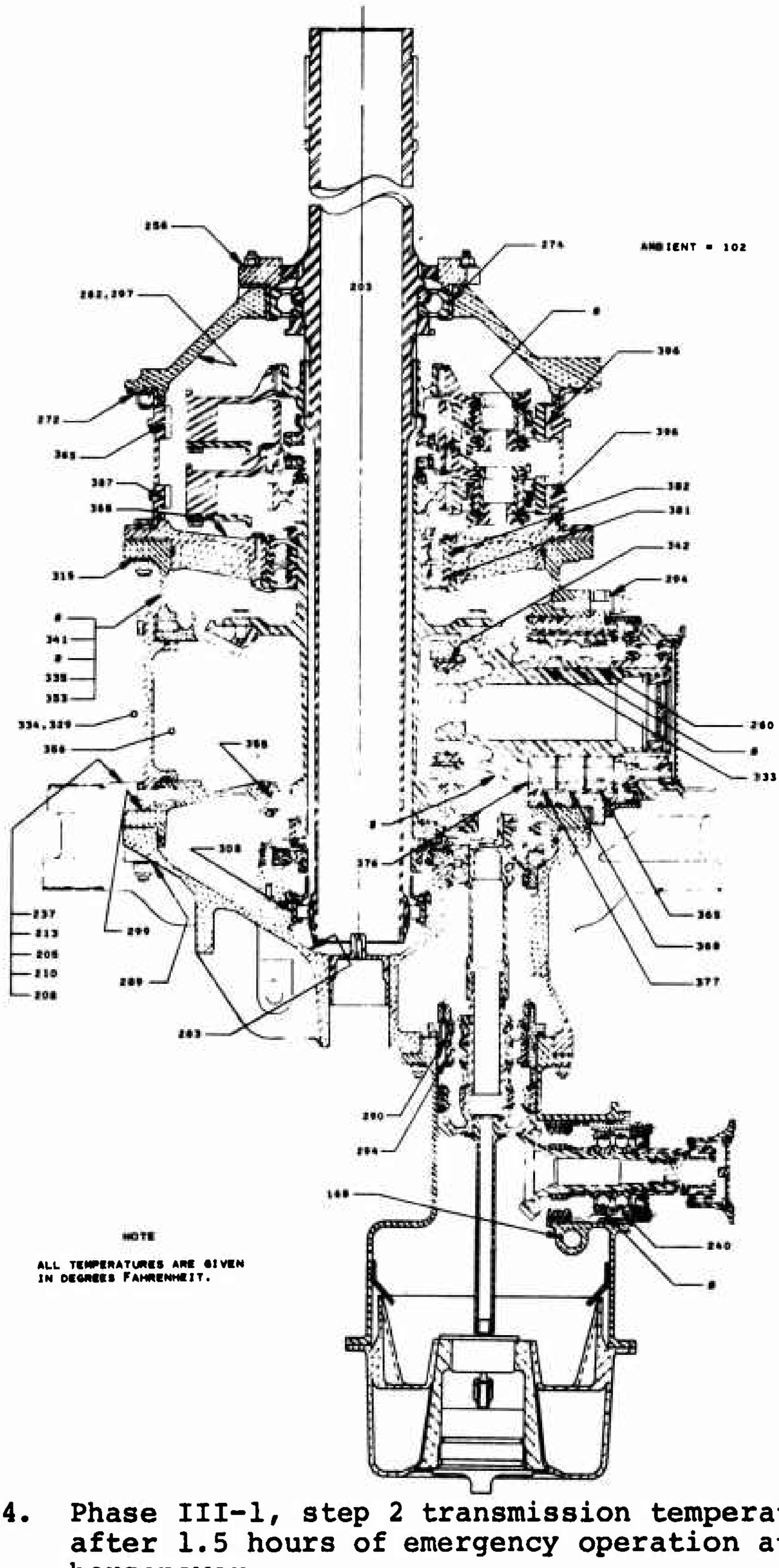

Figure 44. Phase III-1, step 2 transmission temperatures after 1.5 hours of emergency operation at 950 horsepower.

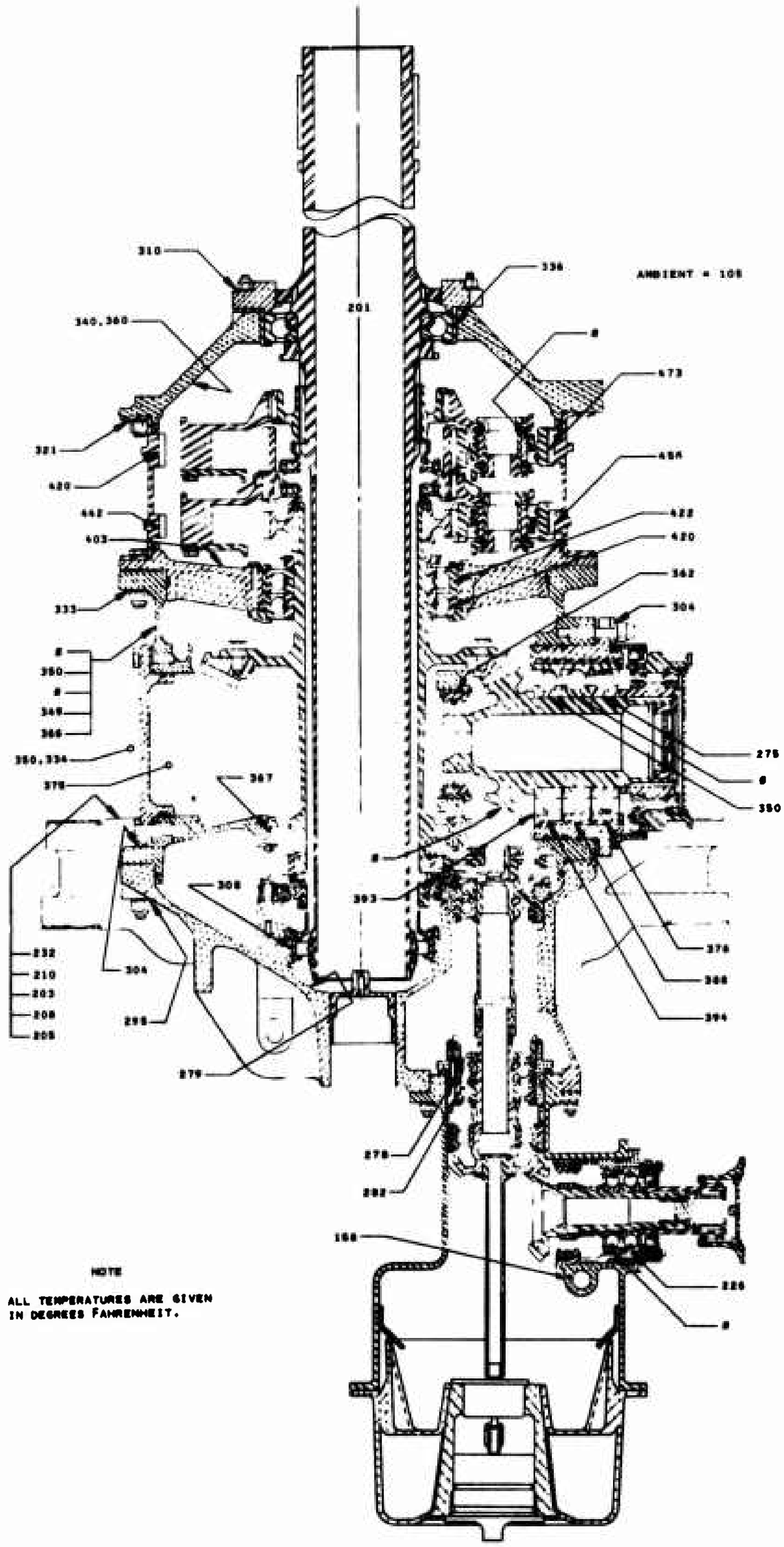

Figure 45. Phase III-1, step 2 transmission temperatures after 2.5 hours of emergency operation at 950 horsepower.

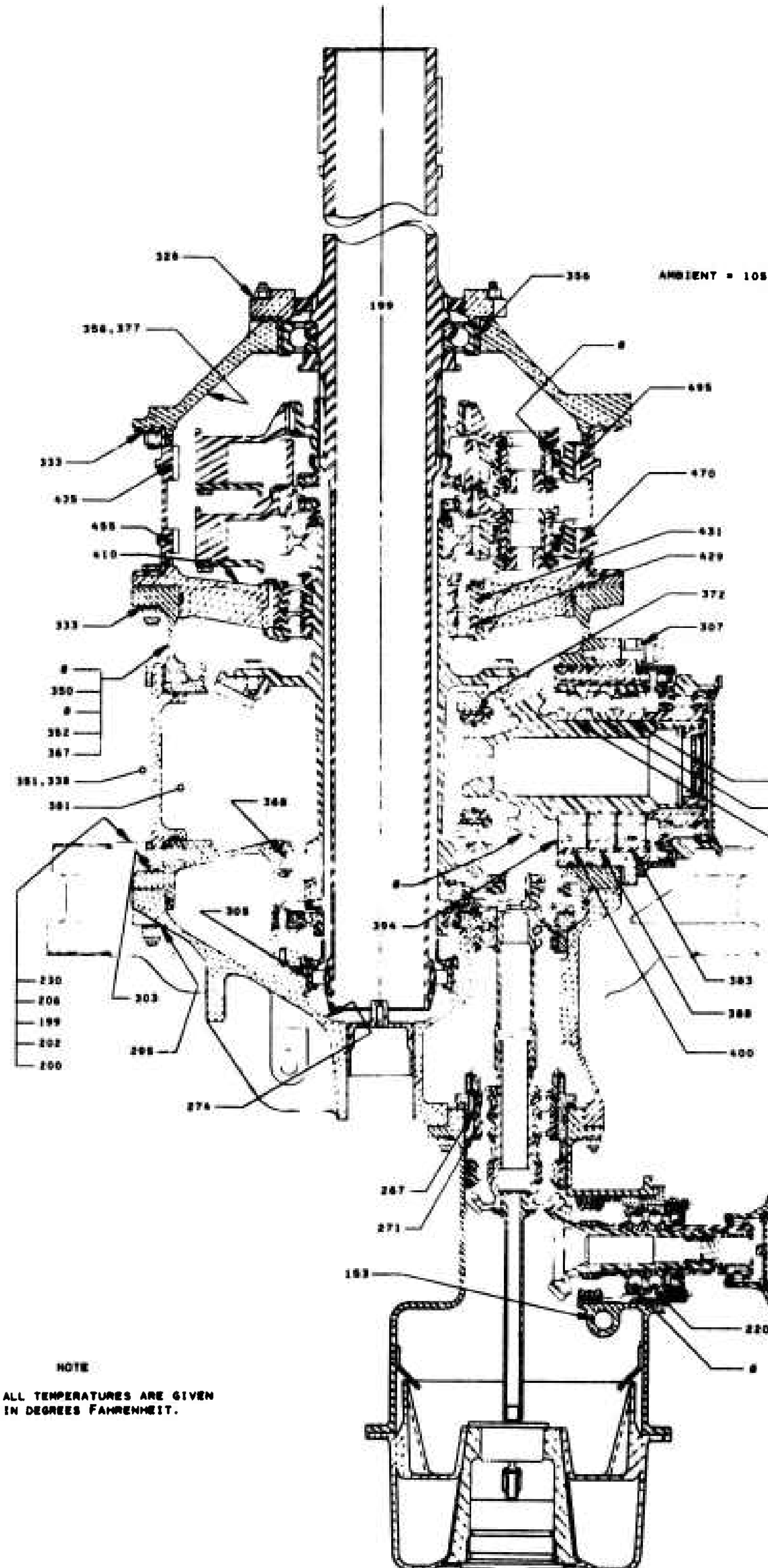

Figure 46. Phase III-1, step 2 transmission temperatures after 3.2 hours of emergency operation at 950 horsepower.

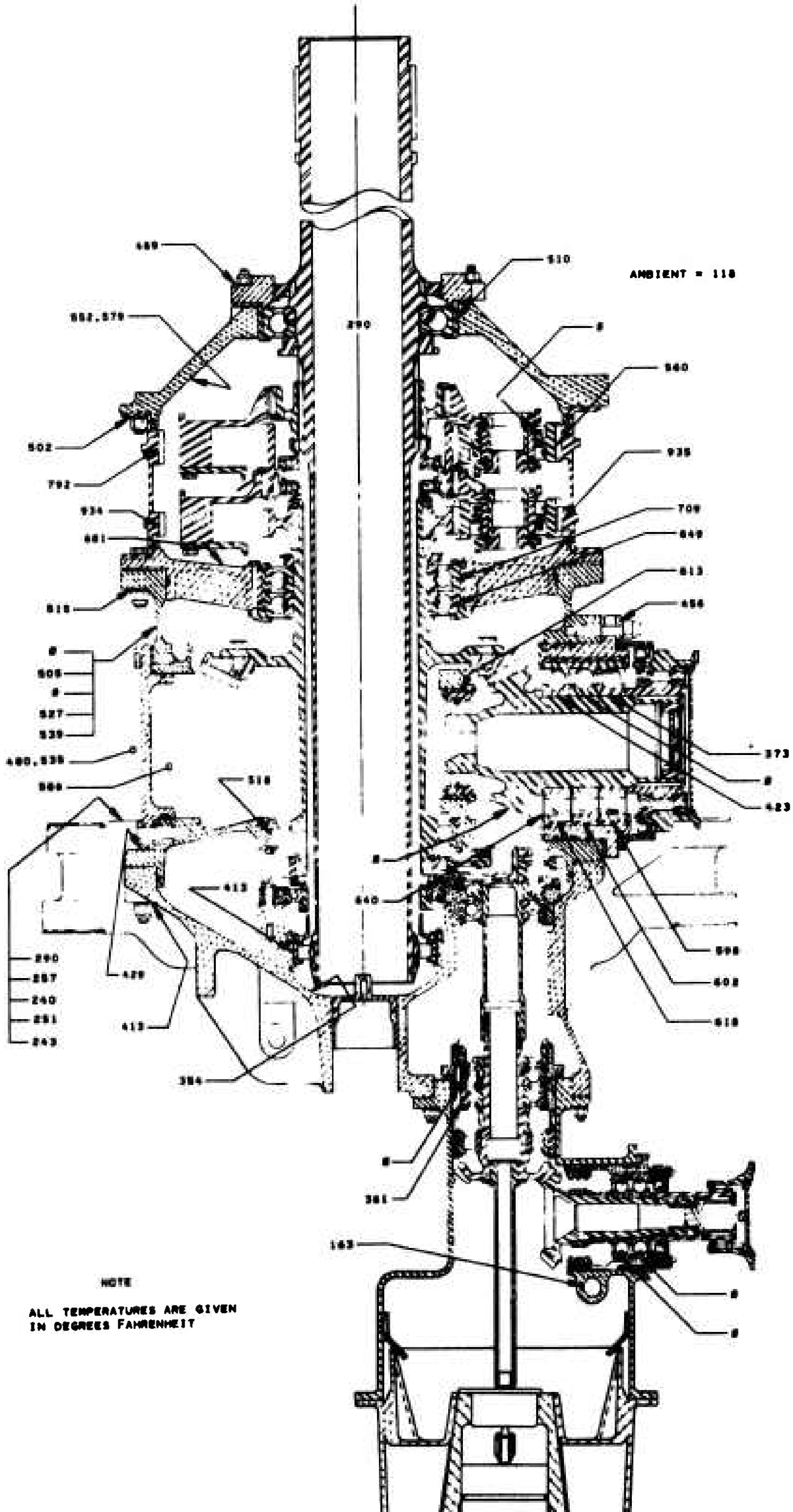

Figure 47. Phase III-1, step 2 transmission temperatures at failure after 4.0 hours of emergency operation at 950 horsepower.

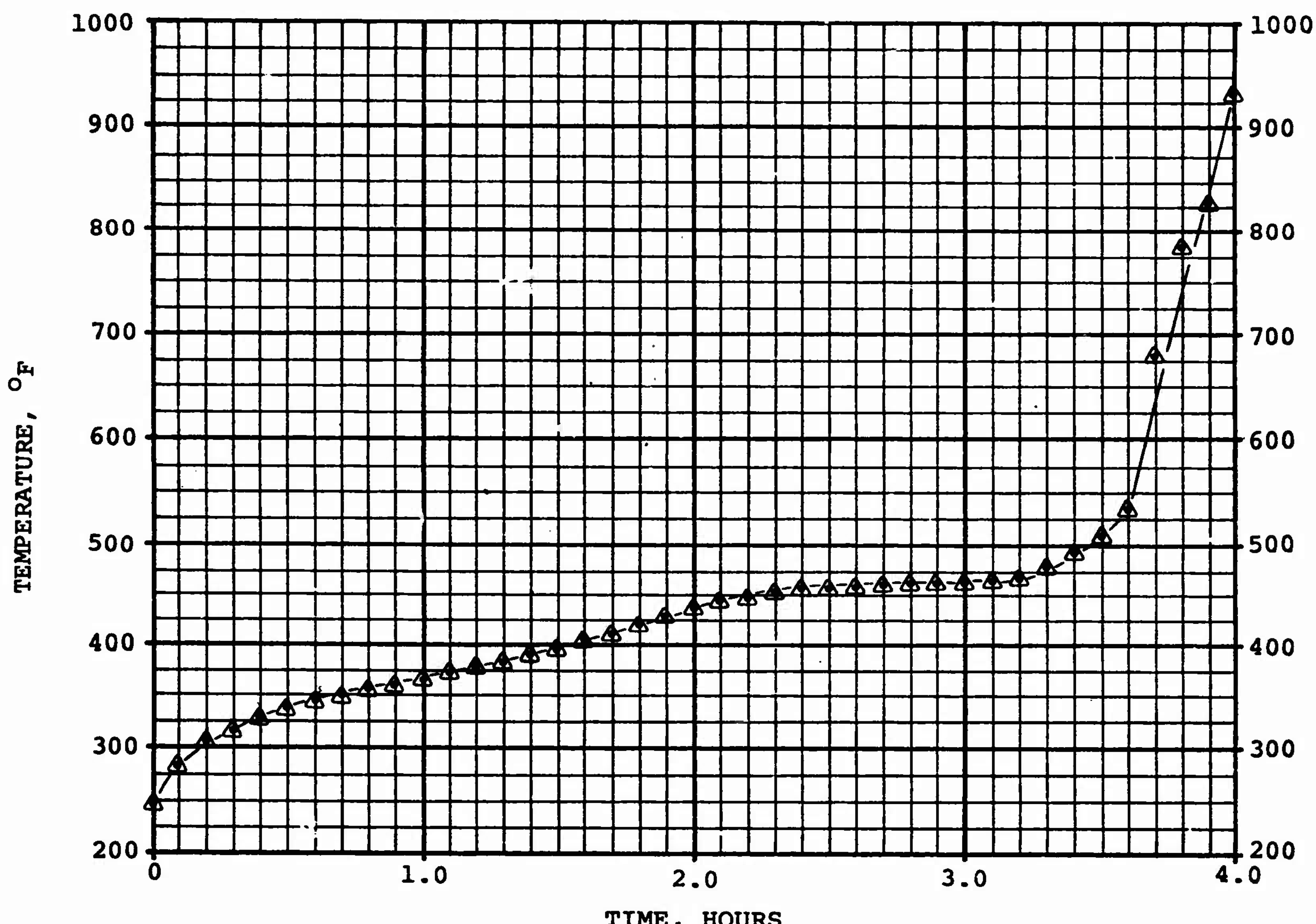

Figure 48. Temperature of lower ring gear mesh (thermocouple 28) during 4.0-hour emergency lubrication run.

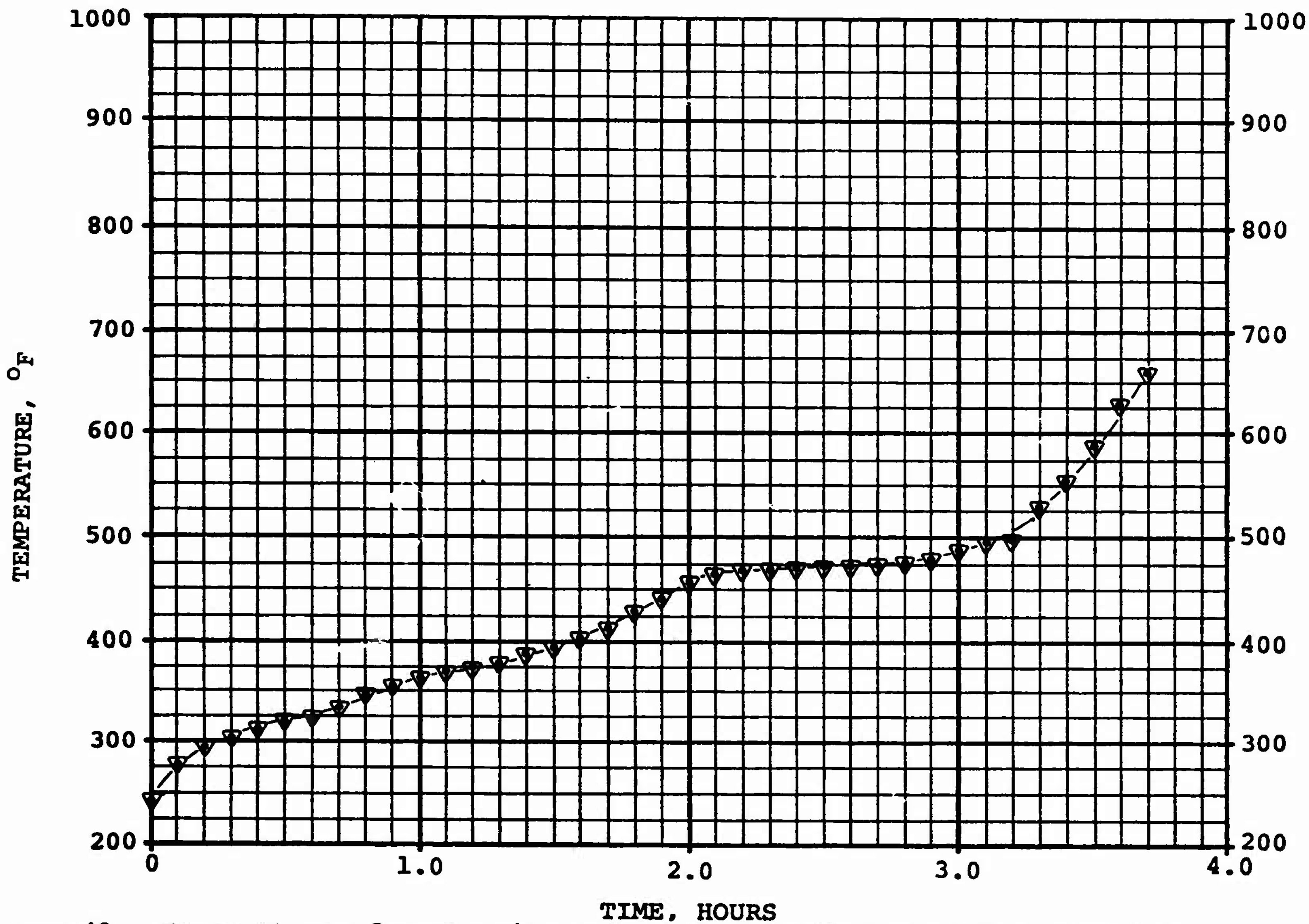

Figure 49. Temperature of upper ring gear mesh (thermocouple 27) during 4.0-hour emergency lubrication run.

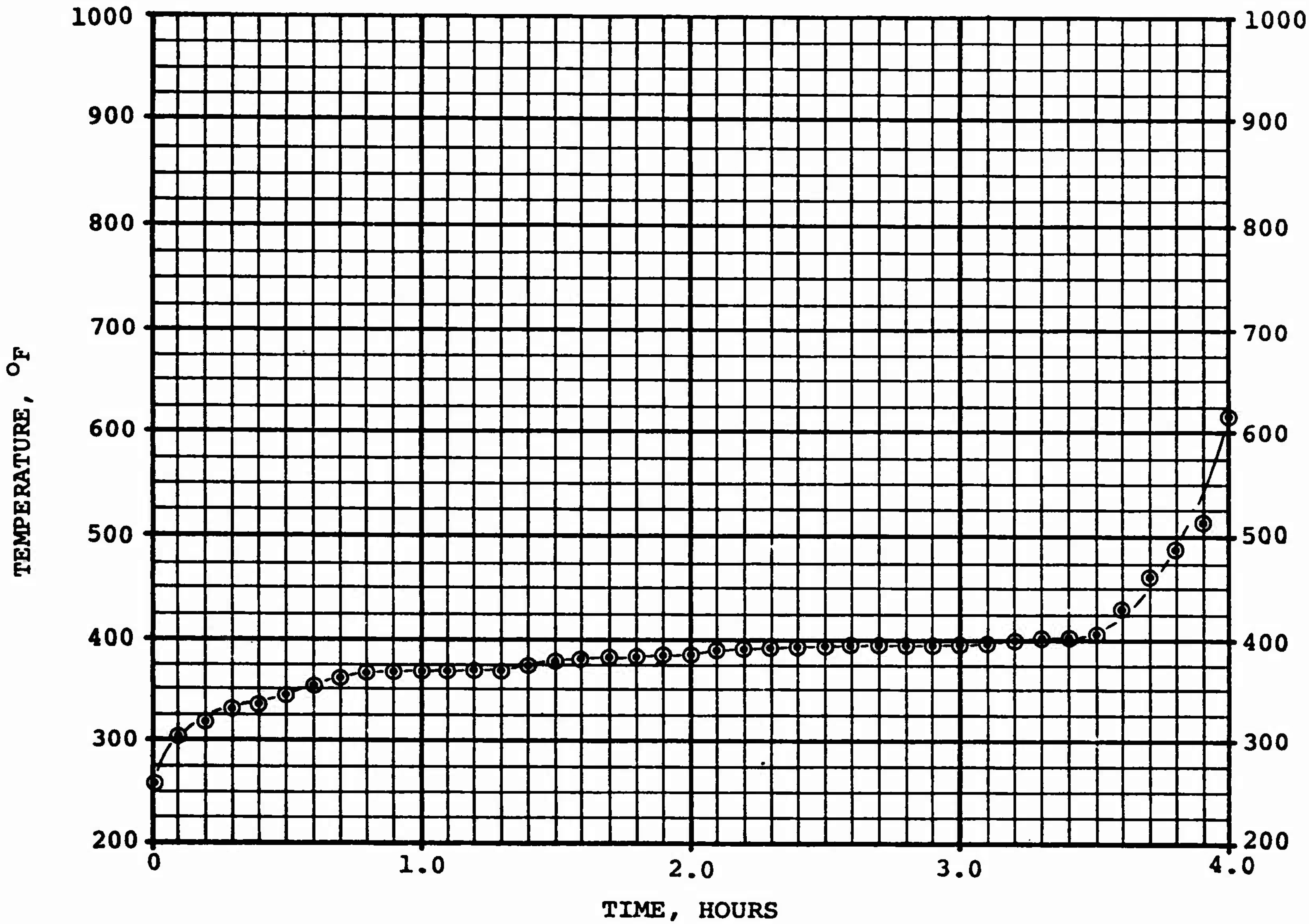

Figure 50. Temperature of inboard triplex bearing, outer ring (thermocouple 34) during 4.0-hour emergency lubrication run.

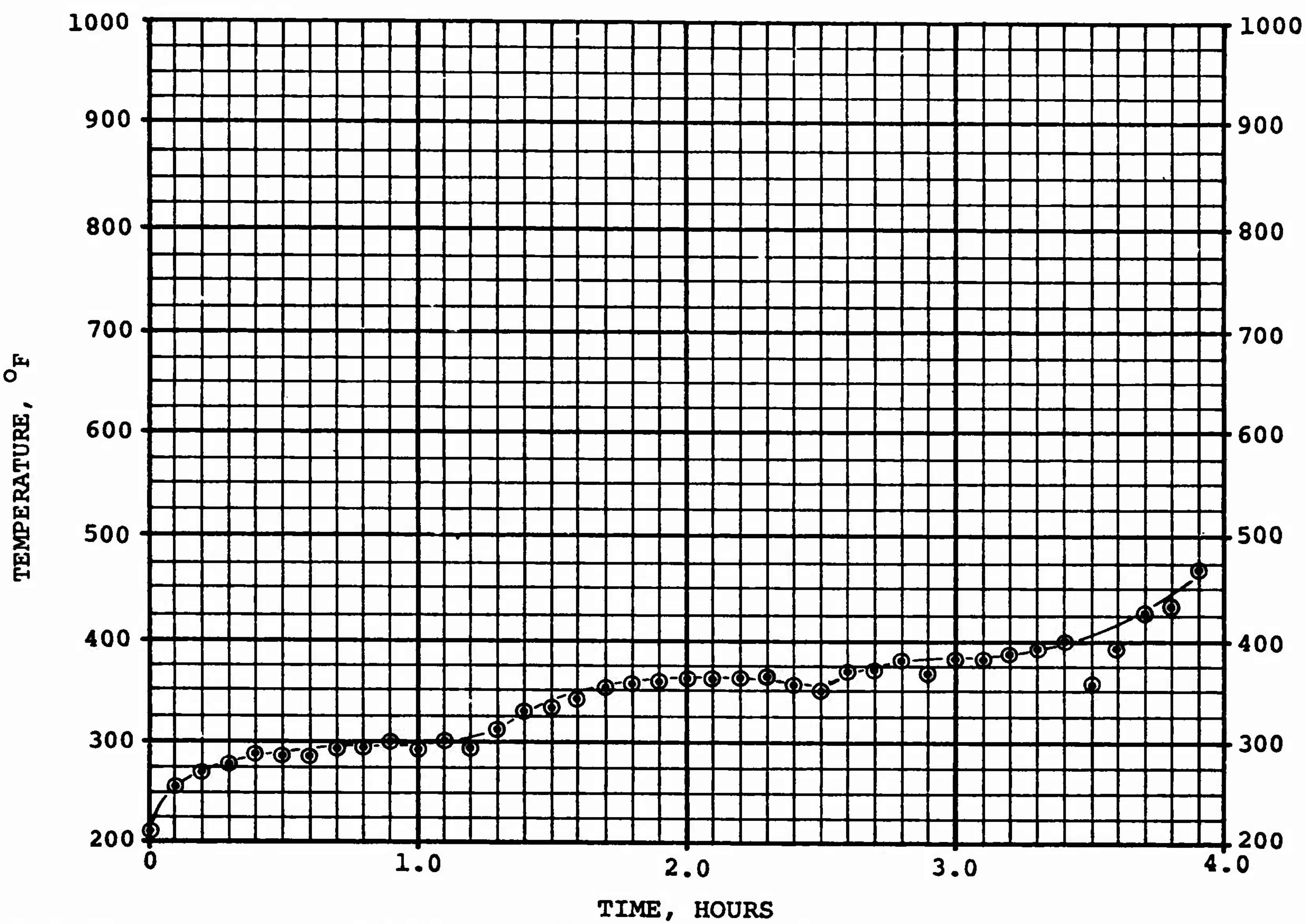

Figure 51. Temperature of inboard triplex bearing, inner ring (thermocouple 38) during 4.0-hour emergency lubrication run.

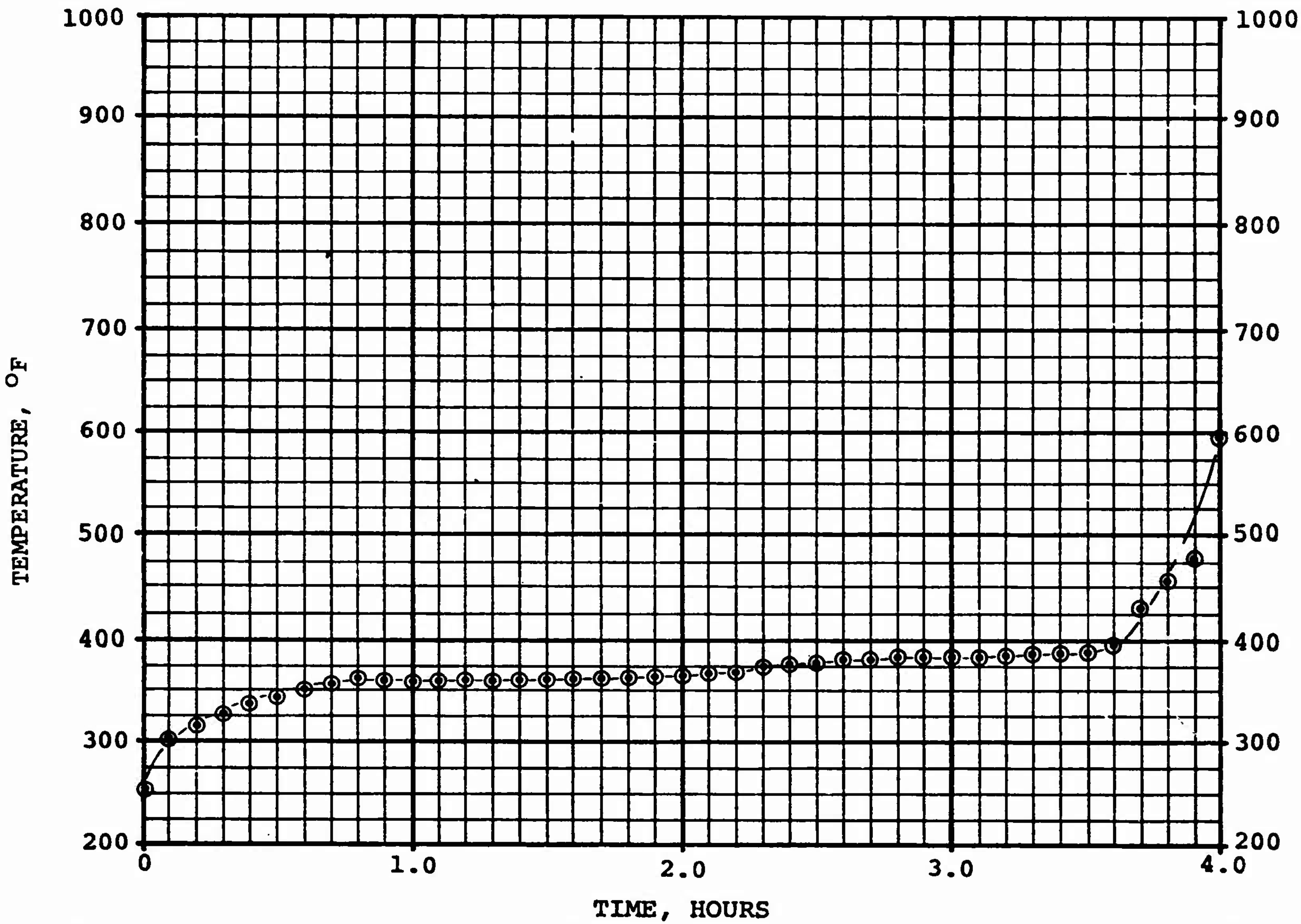

Figure 52. Temperature of outboard triplex bearing, outer ring (thermocouple 36) during 4.0-hour emergency lubrication run.

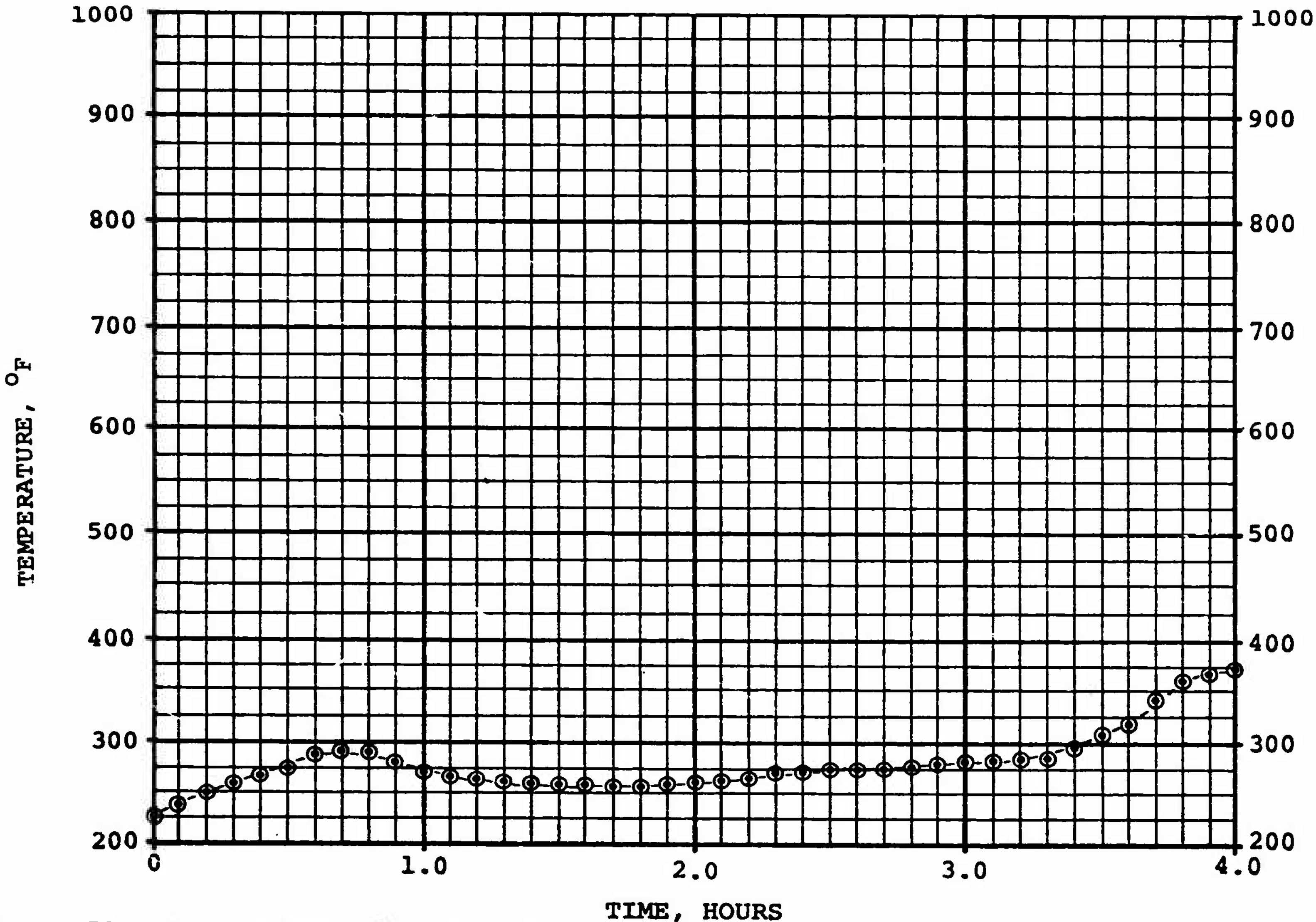

Figure 53. Temperature of outboard triplex bearing, inner ring (thermocouple 40) during 4.0-hour emergency lubrication run.

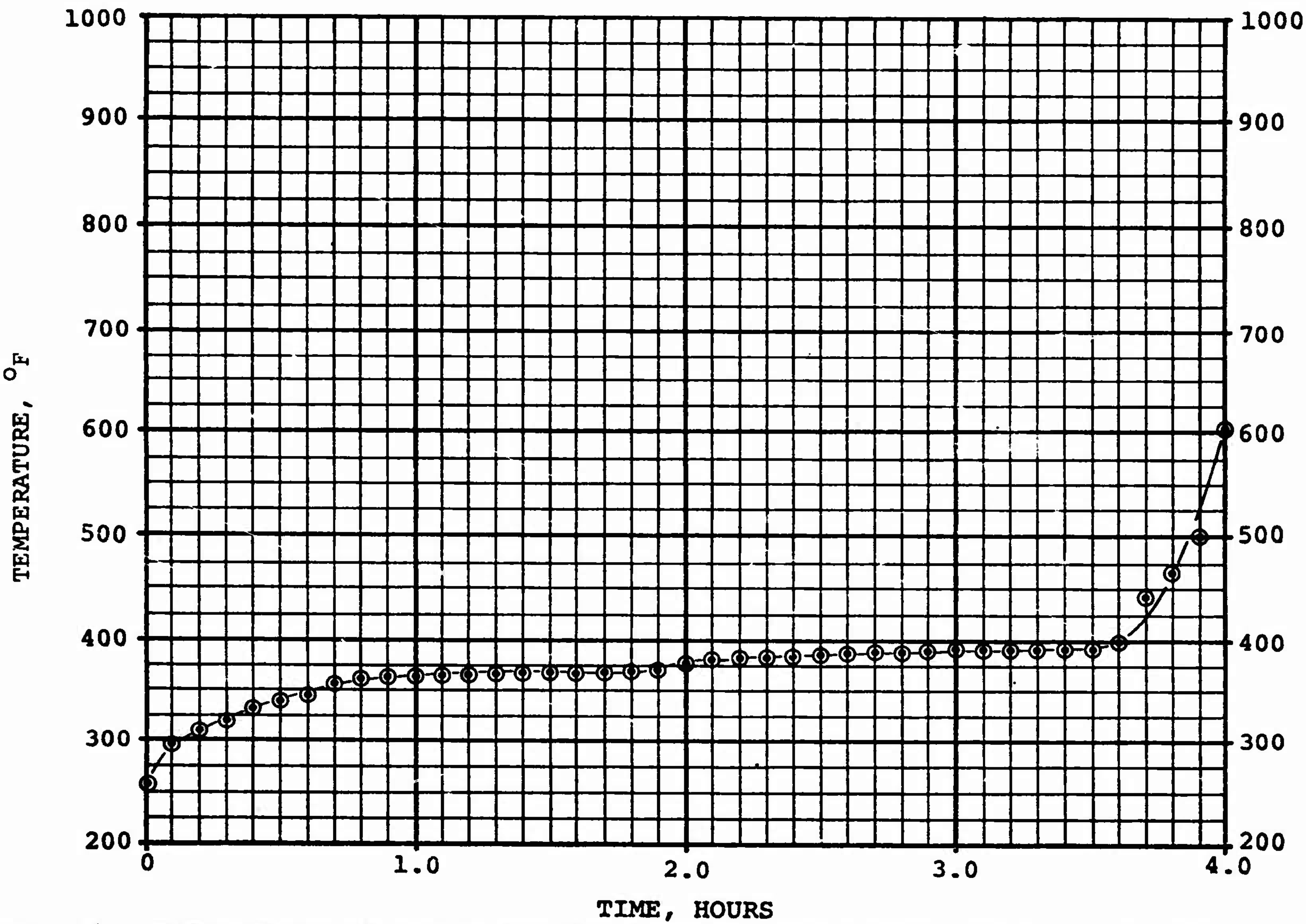

Figure 54. Temperature of center triplex bearing, outer ring (thermocouple 35) during 4.0-hour emergency lubrication run.

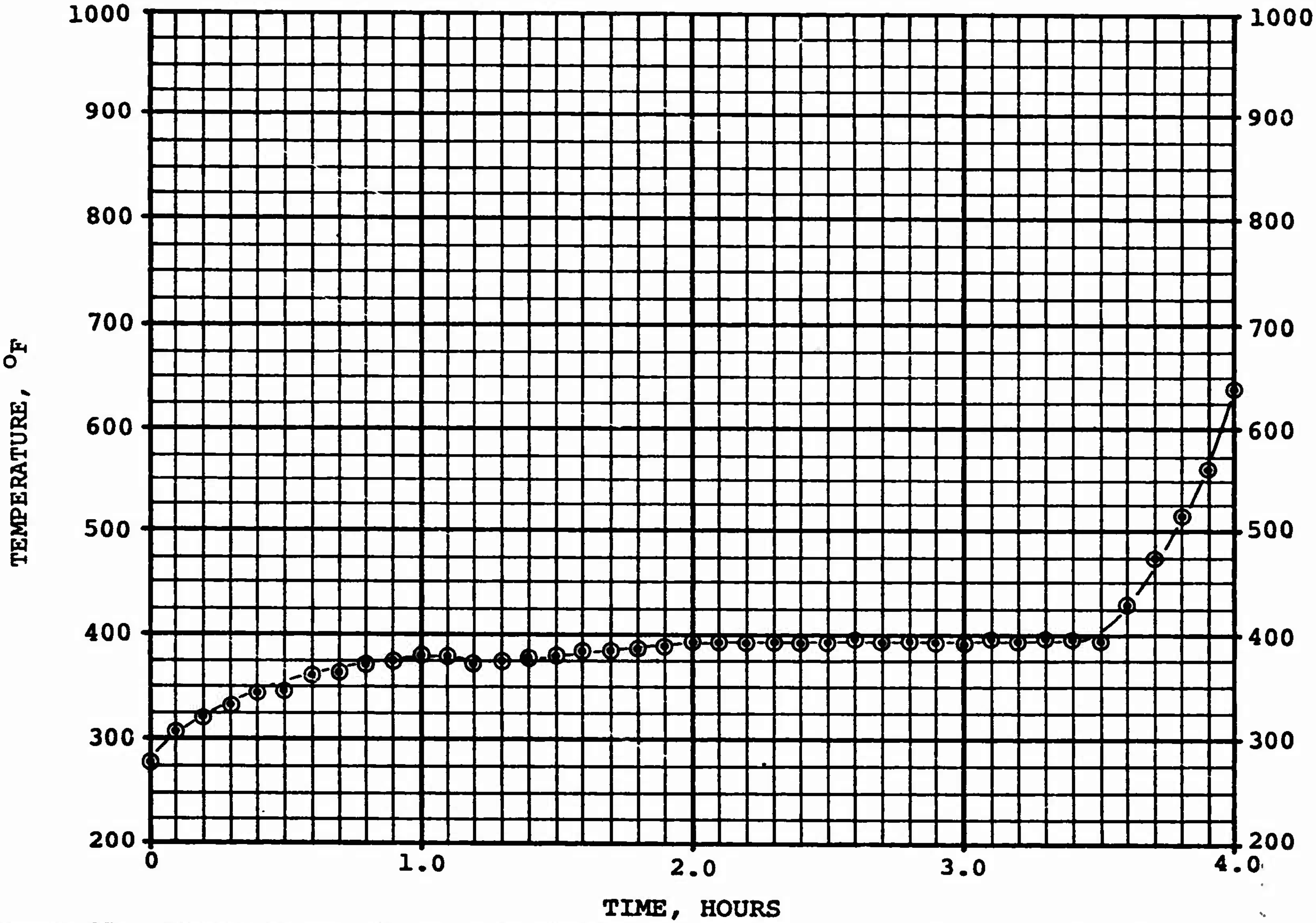

Figure 55. Temperature of oil out of the triplex bearing (thermocouple 41) during the 4.0-hour emergency lubrication run.

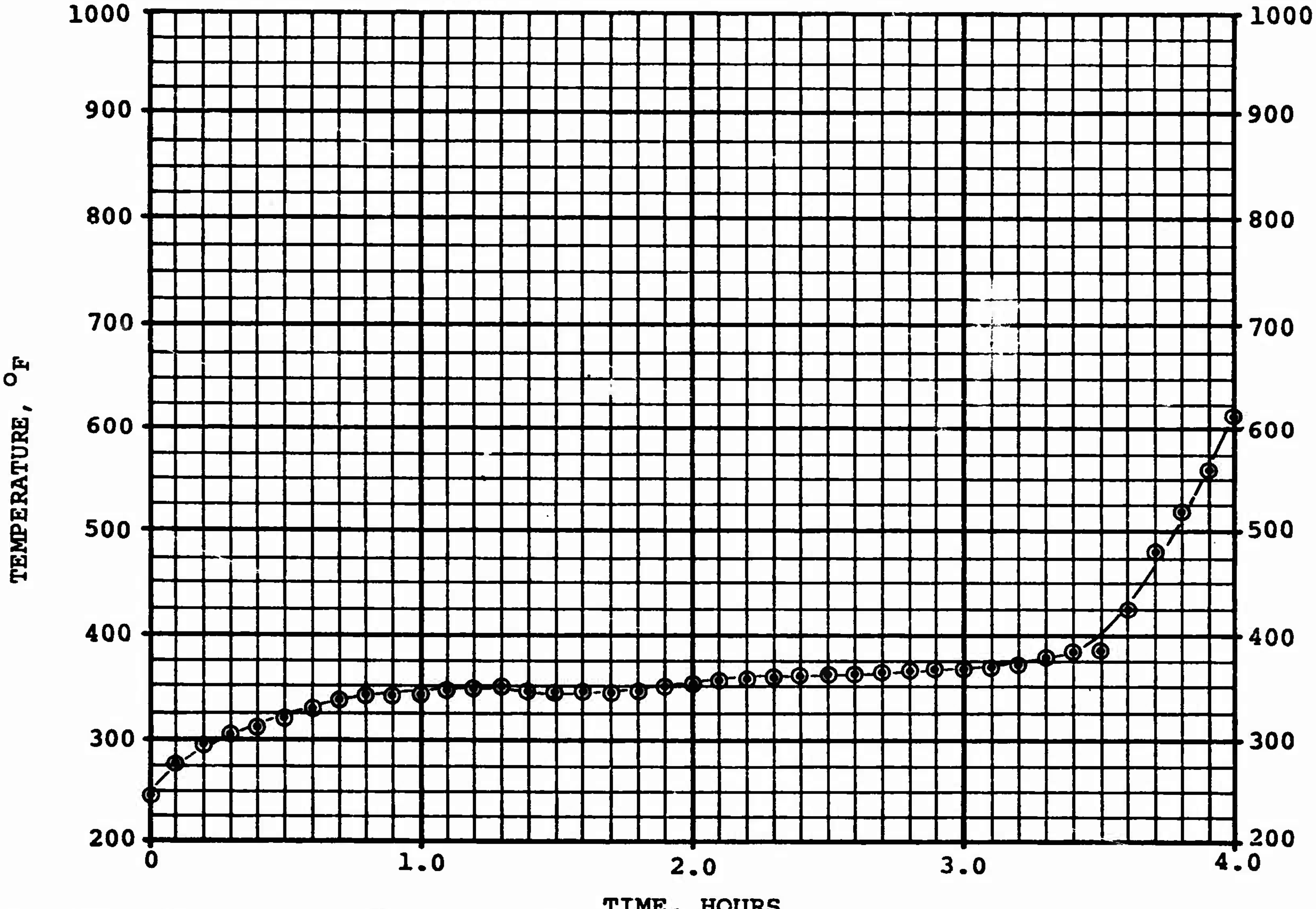

Figure 56. Temperature of input pinion roller bearing, outer ring (thermocouple 7) during 4.0-hour emergency lubrication run.

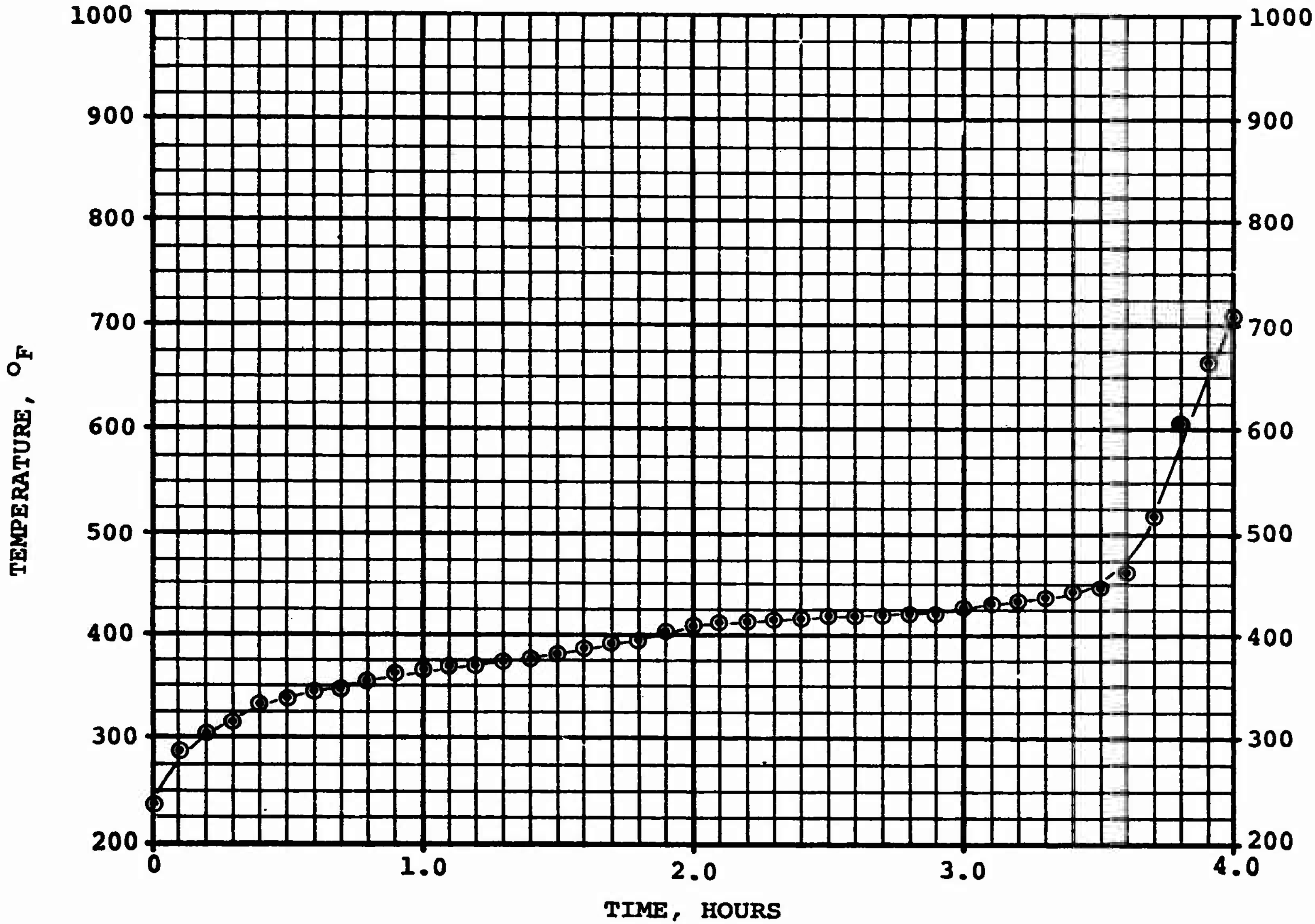

Figure 57. Temperature of bevel gear shaft duplex upper bearing, outer ring (thermocouple 5) during 4.0-hour emergency lubrication run.

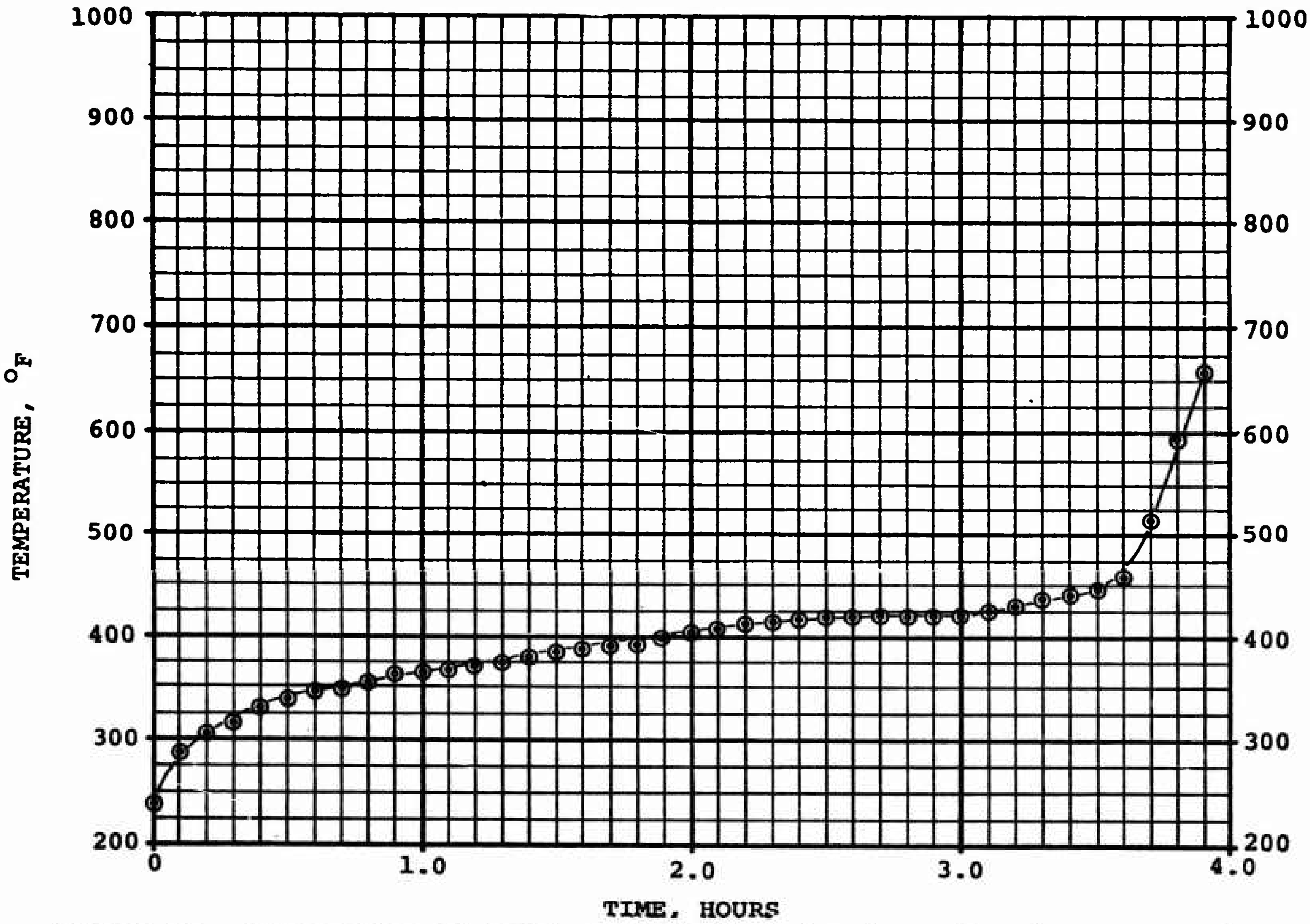

Figure 58. Temperature of bevel gear shaft duplex lower bearing, outer ring (thermocouple 6) during 4.0-hour emergency lubrication run.

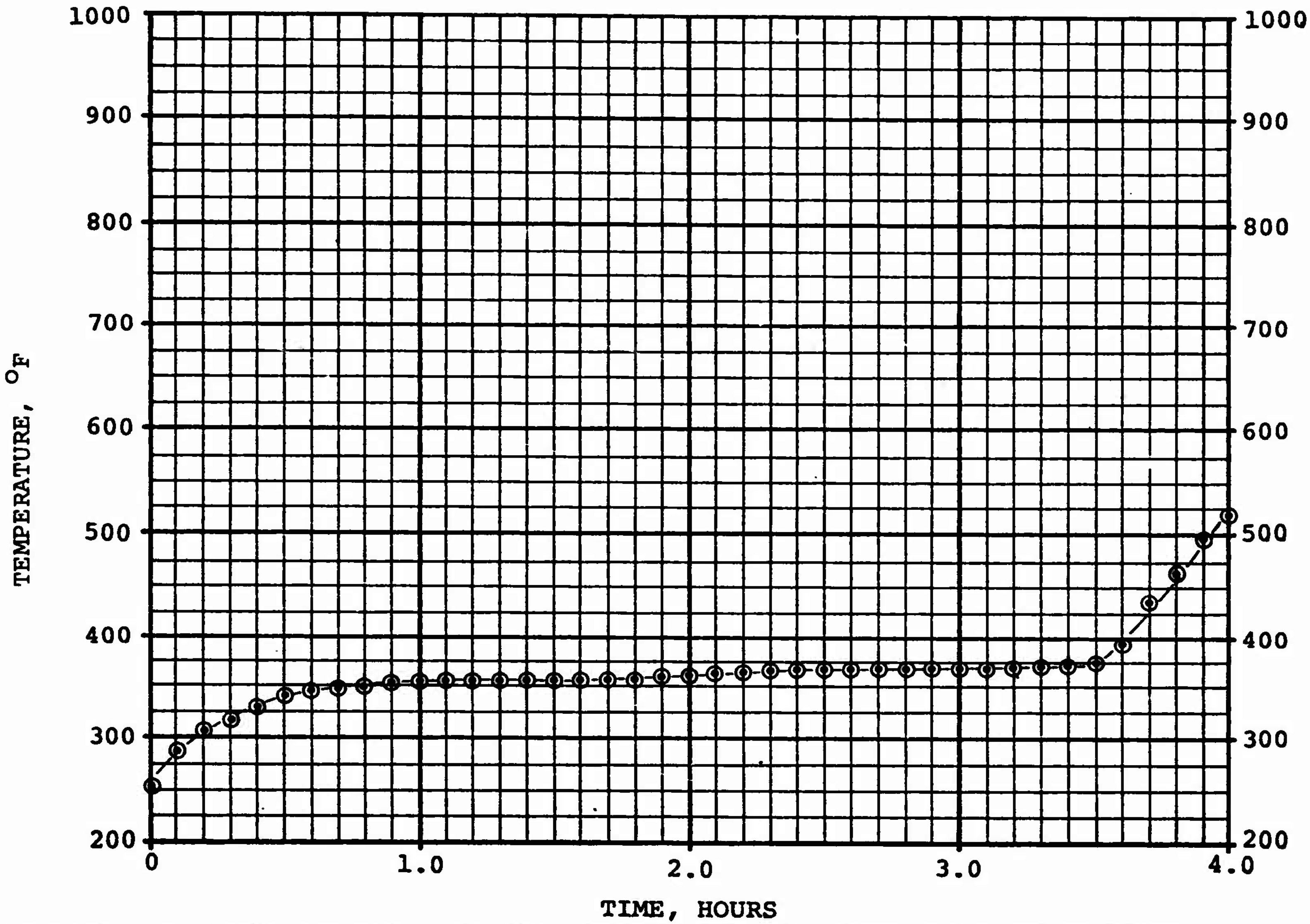

Figure 59. Temperature of bevel gear shaft roller bearing, outer ring (thermocouple 8) during 4.0-hour emergency lubrication run.

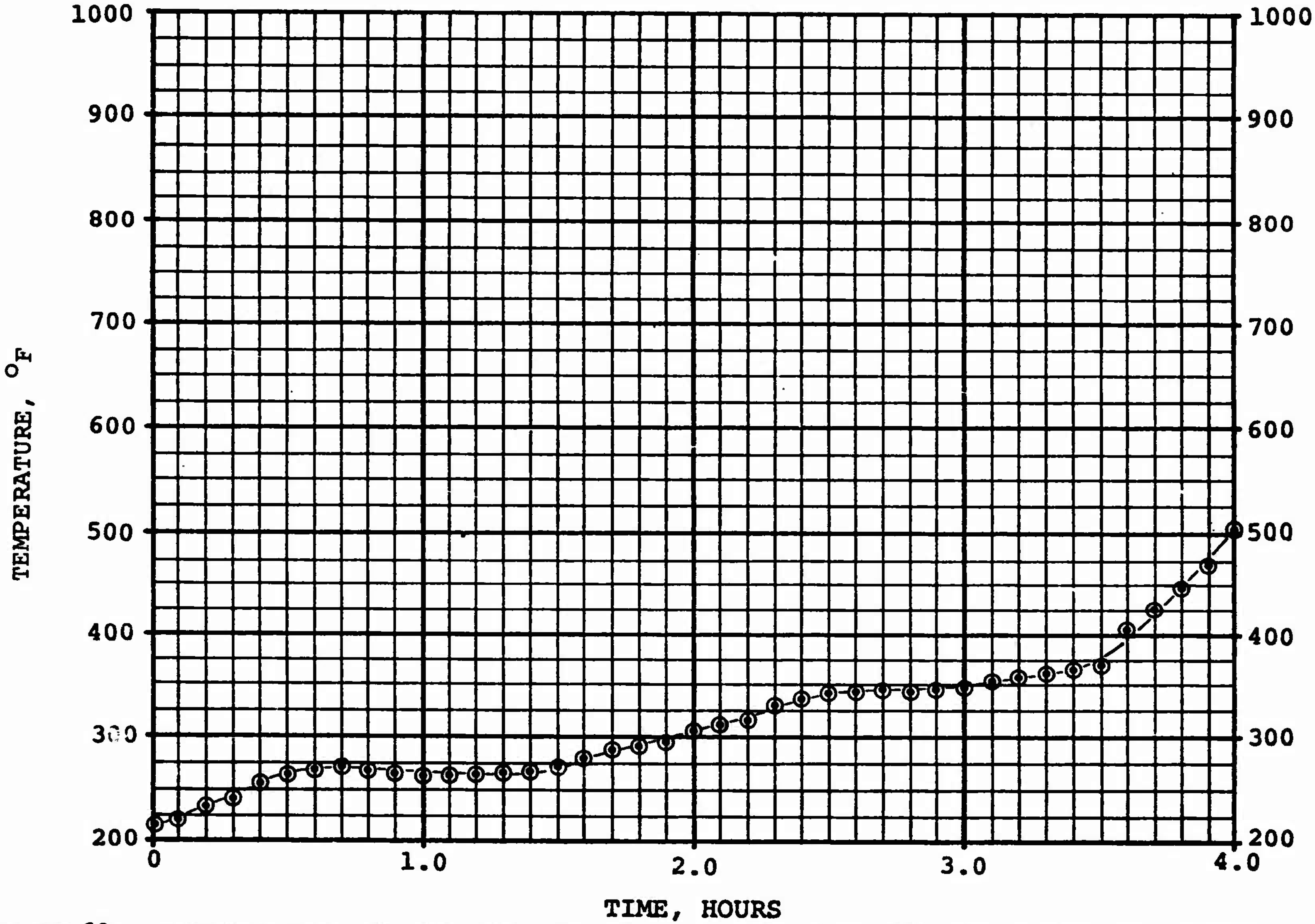

Figure 60. Temperature of mast ball bearing, outer ring (thermocouple 57) during 4.0-hour emergency lubrication run.

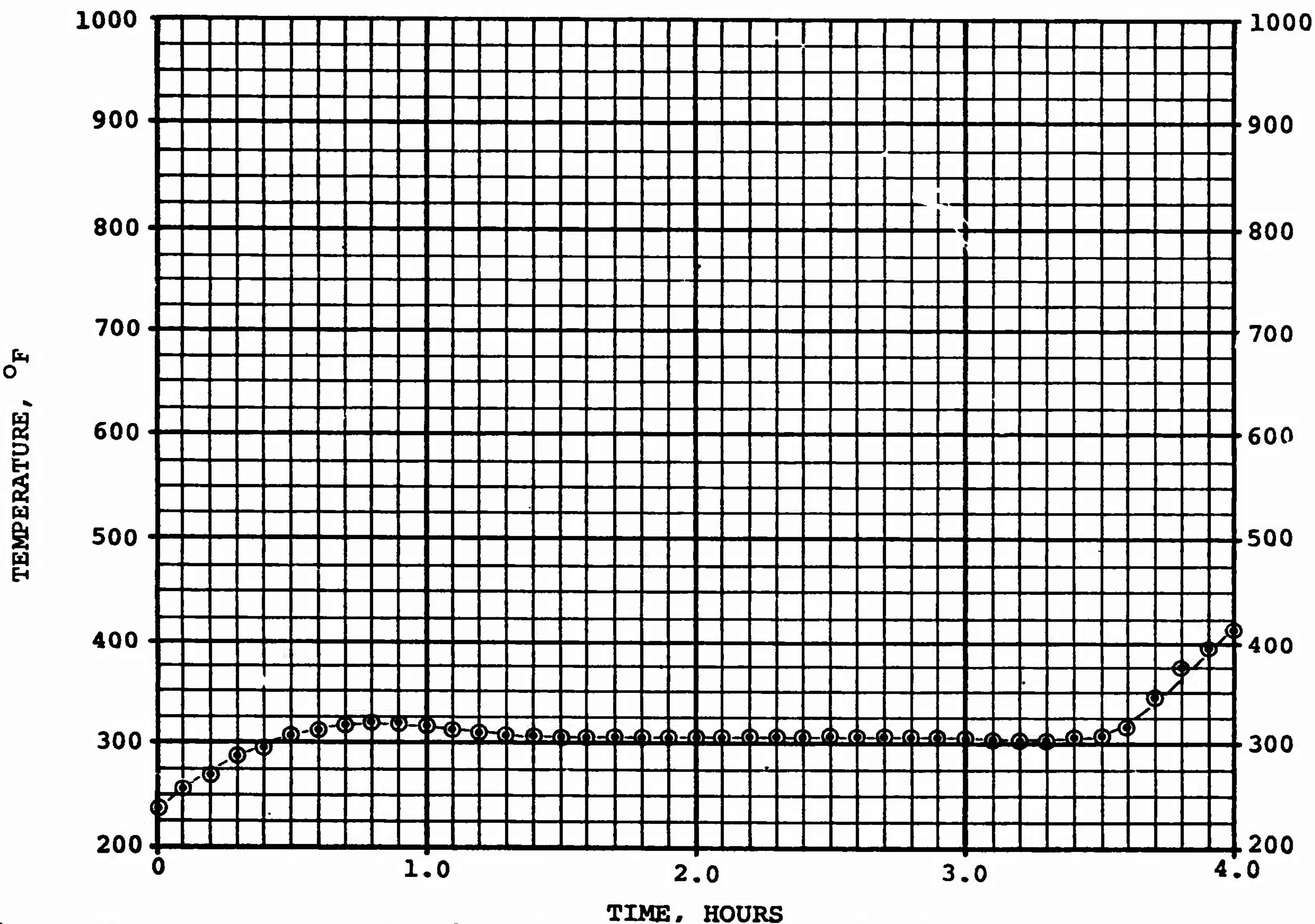

Figure 61. Temperature of mast roller bearing, outer ring (thermocouple 56) during 4.0-hour emergency lubrication run.

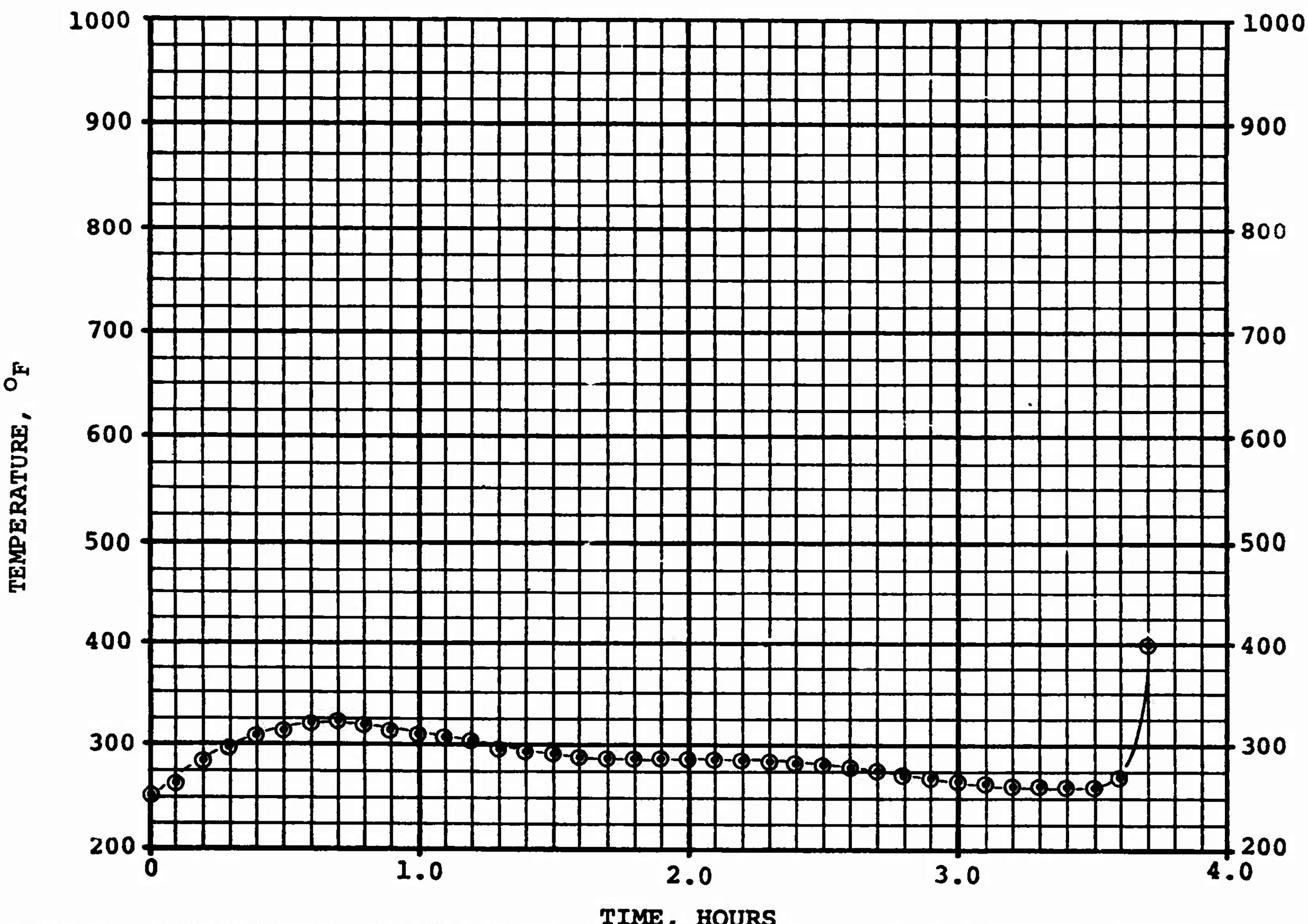

Figure 62. Temperature of tail rotor drive input duplex upper bearing, outer ring (thermocouple 31) during 4.0-hour emergency lubrication run.

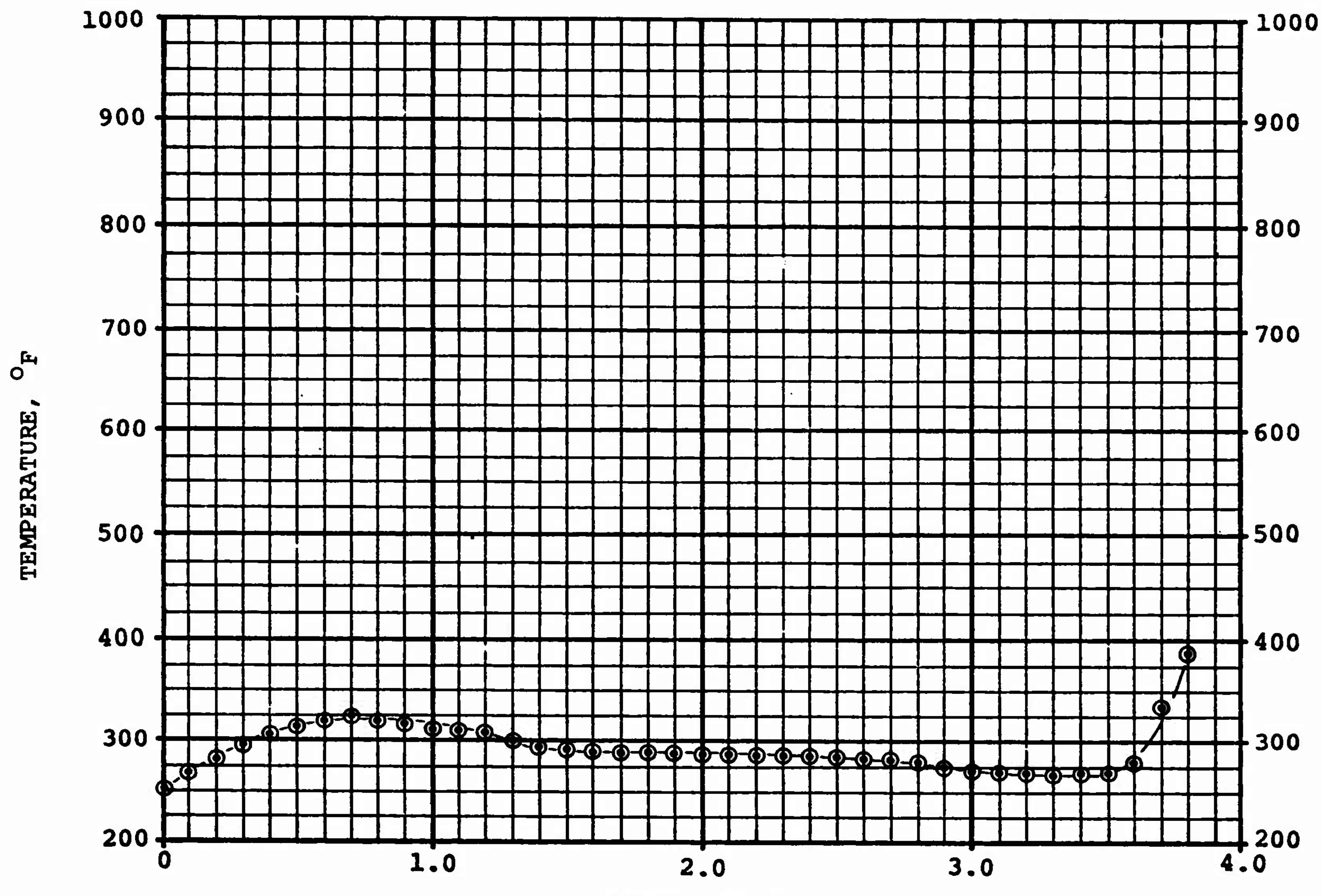

Figure 63. Temperature of tail rotor drive input duplex lower bearing, outer ring (thermocouple 42) during 4.0-hour emergency lubrication run.

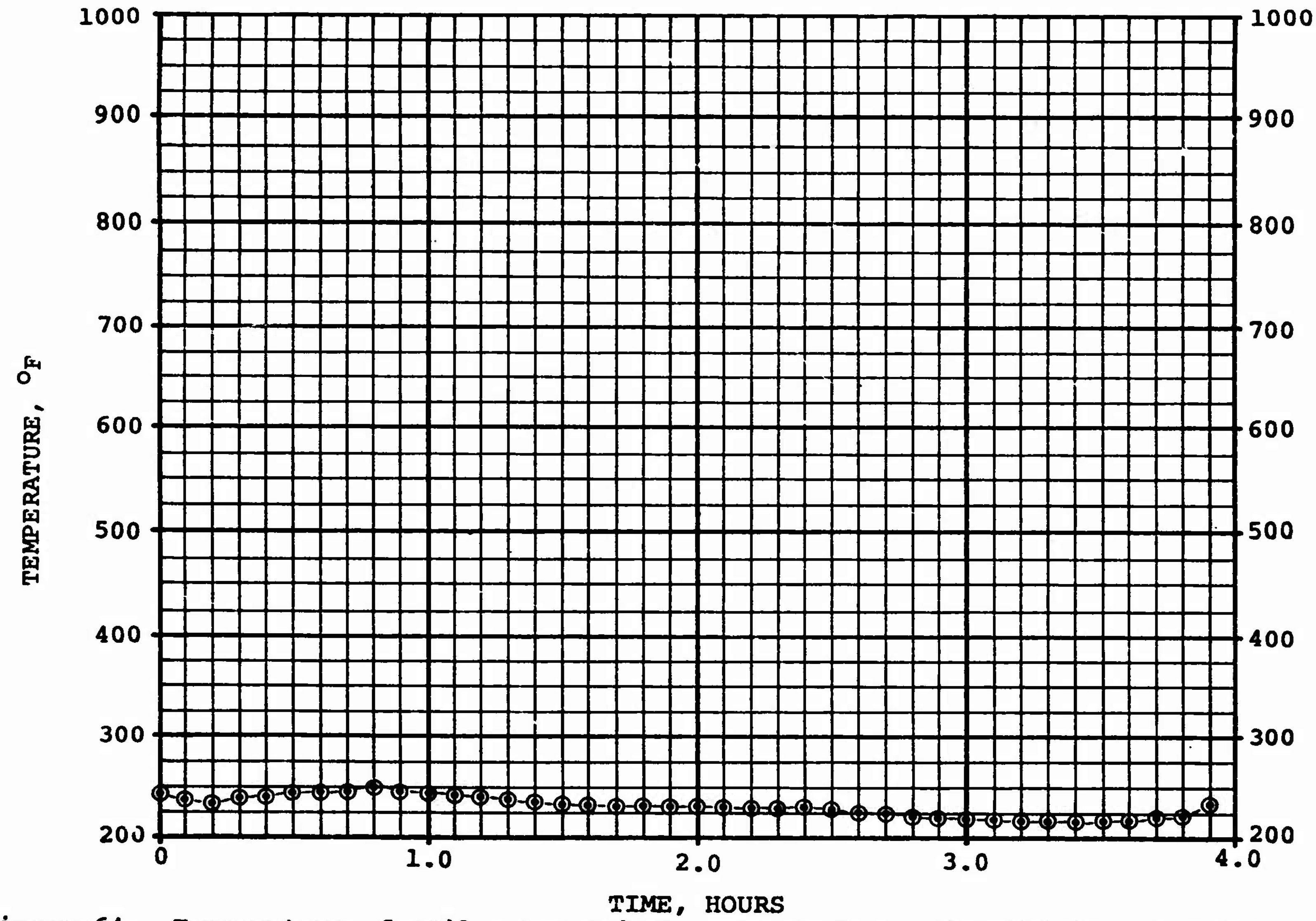

Figure 64. Temperature of tail rotor drive output duplex outboard bearing, outer ring (thermocouple 32) during 4.0-hour emergency lubrication run.

5.4.5 Detailed Results of Visual Post-Run Inspection

5.4.5.1 Upper Mast Ball Bearing

This bearing turned freely and was still oily when removed from the transmission. The balls were not discolored. The races and the retainer appeared to be in excellent condition.

5.4.5.2 Upper Planetary Assembly

This assembly was still functional when removed from the transmission and is shown in Figure 65. All components were dry.

- The upper planet pinion was spattered with coke deposits but otherwise appeared unharmed, as shown in Figure 66.

- The upper planetary rollers were discolored but appeared to be unharmed, as shown in Figure 66.

- The upper planetary retainers appeared to be in good condition, as shown in Figure 66. The silver plating was still complete and appeared to have been unaffected by the high-temperature operation.

- The upper planetary thrust washers appeared unharmed and are shown in Figure 66.

5.4.5.3 Upper Planetary Support Bearing

This bearing still turned and was functional; however, it appeared that the 52100 races had been deformed at the high temperatures. The bearing did not turn smoothly.

5.4.5.4 Upper Sun Gear

At first glance it appeared that the sun planet mesh had lost clearance, since there was a pattern on both sides of the sun gear teeth; however, further investigation (scotch brite and magnification) revealed that neither side of the teeth was damaged, and the coast side pattern was typical of any observed on production (used) sun gears. Figure 67 shows the sun gear as removed from the transmission.

5.4.5.5 Upper Ring Gear

The upper ring gear exhibited no visual damage, as can be seen in Figure 68.

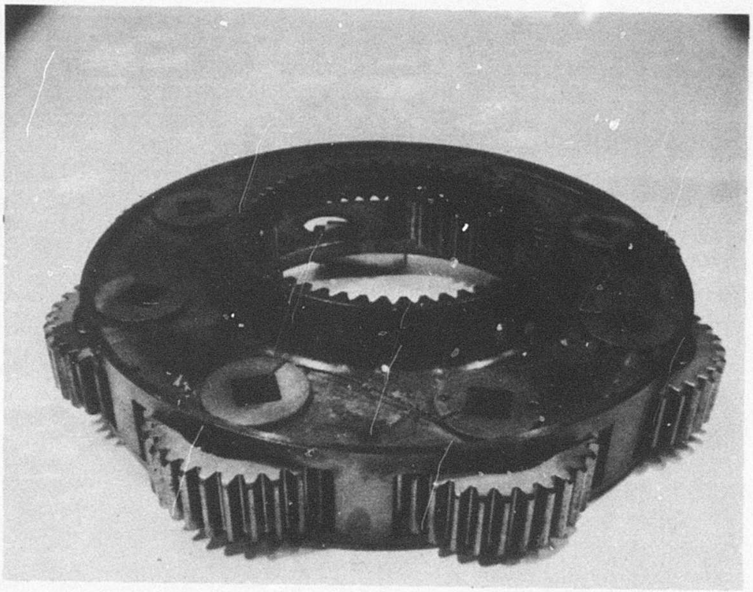

Figure 65. Upper planetary assembly following 4.0-hour emergency lubrication run.

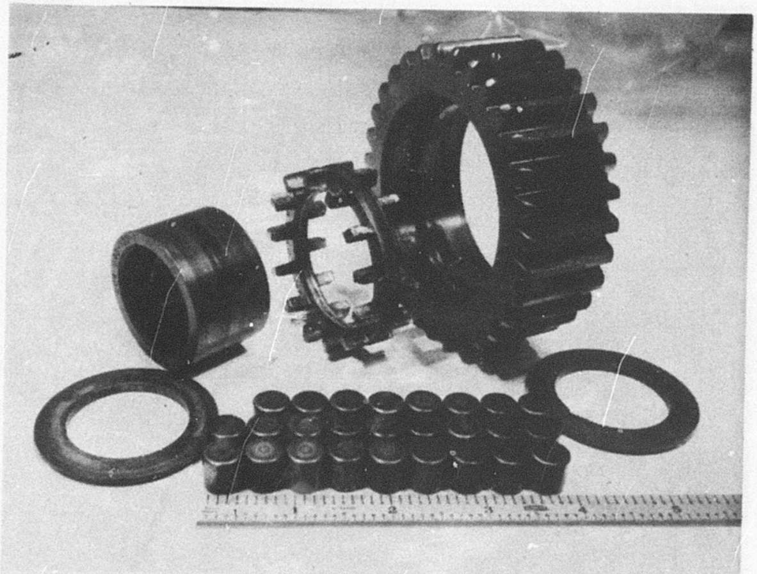

Figure 66. Upper planetary pinion, retainer, thrust washers, rollers, and races following 4.0-hour emergency lubrication run.

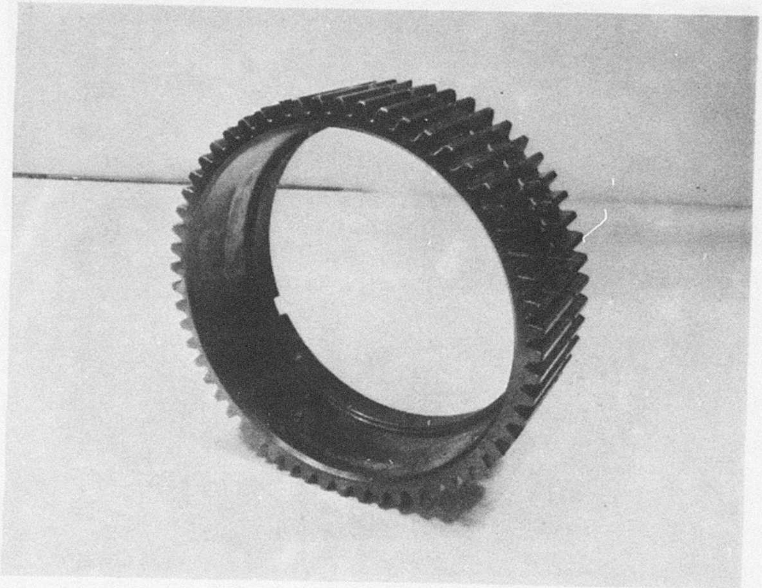

Figure 67. Upper sun gear following 4.0-hour emergency lubrication run.

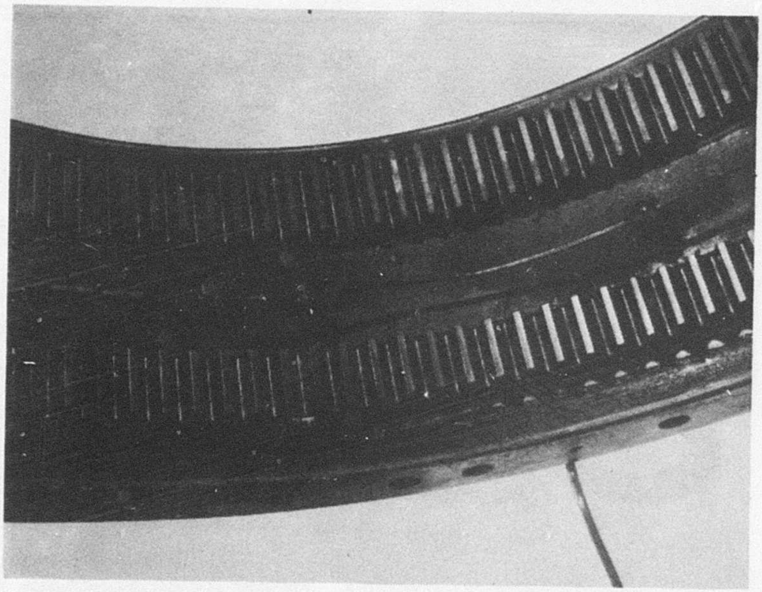

Figure 68. Upper and lower ring gear following 4.0-hour emergency lubrication run.

5.4.5.6 Lower Sun Gear

The lower sun gear failed. No external teeth remained on the failed gear, as shown in Figure 69.

5.4.5.7 Lower Ring Gear

The lower ring gear appeared to be unharmed, as shown in Figure 68.

5.4.5.8 Lower Planetary Assembly

All components of this assembly were dry when removed from the transmission. Figure 70 shows the removed assembly.

- Figure 71 shows the lower planetary pinion teeth, which appeared to have been softened and deformed at the high temperatures and then reformed by the ring gear following complete failure of the lower sun gear. The inside diameter of the pinion which forms the outer race for the planetary rollers had about a .002-inch step where the M-50 rollers had been running. The inner races of the planetary rollers also had a step, but it was only slightly discernible.

- The lower planetary rollers were discolored but appeared unharmed, as shown in Figure 71.

- The lower planetary retainers, shown in Figure 71, appeared to be undamaged. The silver plating was still complete and appeared to be unaffected by the high temperature operation.

- Figure 71 shows the lower planetary thrust washers, which appeared to be unharmed.

5.4.5.9 Lower Planetary Support Bearing

The bearing still turned and was functional; however, it appeared that the 52100 races had been deformed at the high temperatures. The bearing did not turn smoothly.

5.4.5.10 Input Gear Shaft Duplex Bearing

The duplex bearing turned tightly while it was still installed in the 204-040-362 Quill Assembly. When the bearing was removed, it turned freely and disassembly of the bearing itself revealed no visual damage other than discoloration of the balls. The -362 Quill Assembly was probably warped as a result of the high-temperature operation. Figure 72 depicts this duplex bearing.

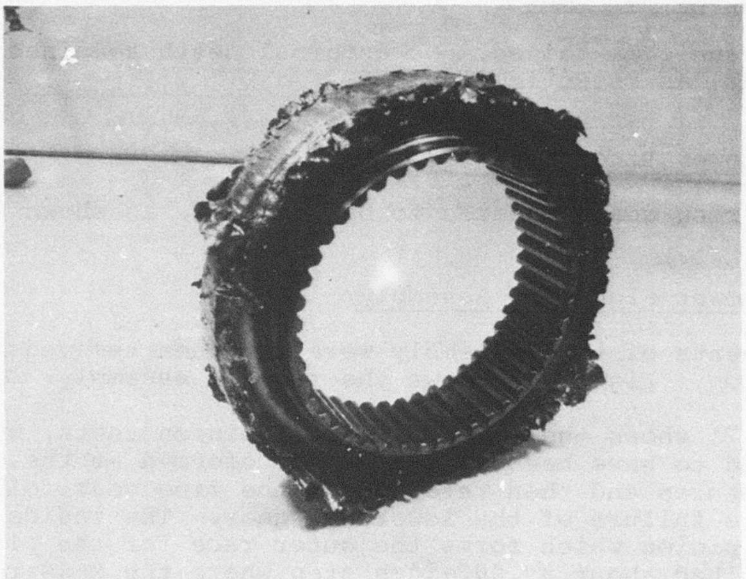

Figure 69. Lower sun gear following 4.0-hour emergency lubrication run.

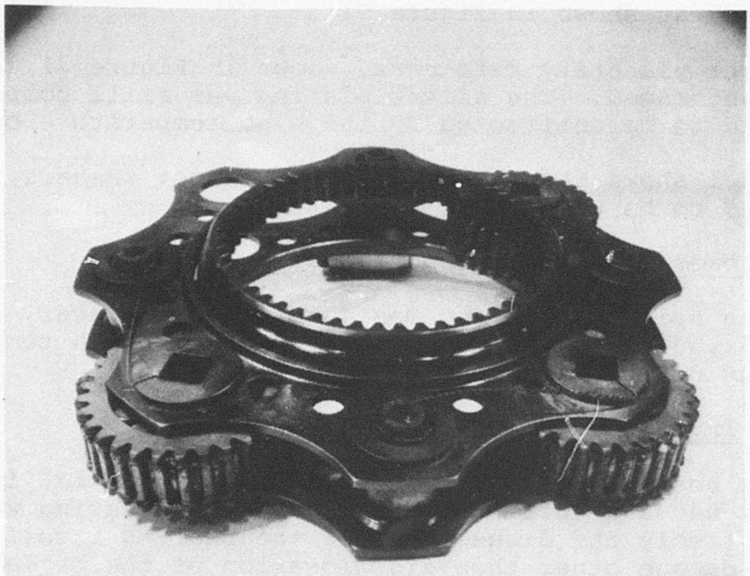

Figure 70. Lower planetary assembly following 4.0-hour emergency lubrication run.

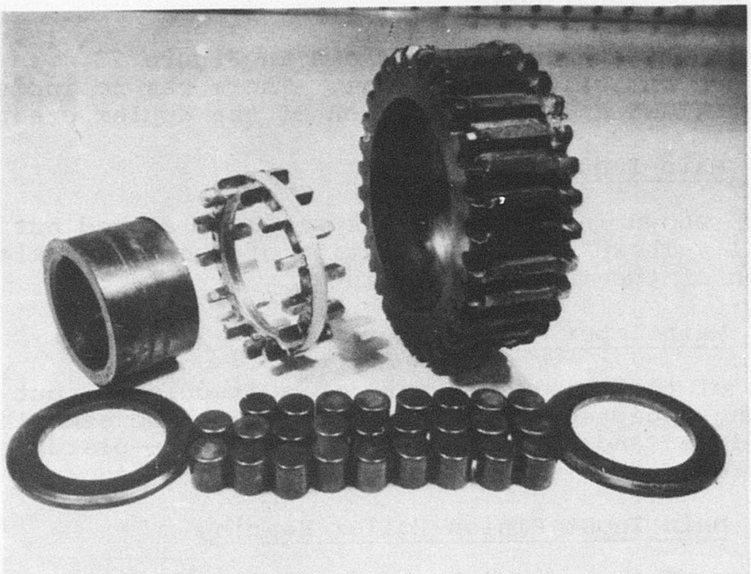

Figure 71. Lower planetary pinion, retainer, thrust washers, rollers, and races for 4.0-hour emergency lubrication run.

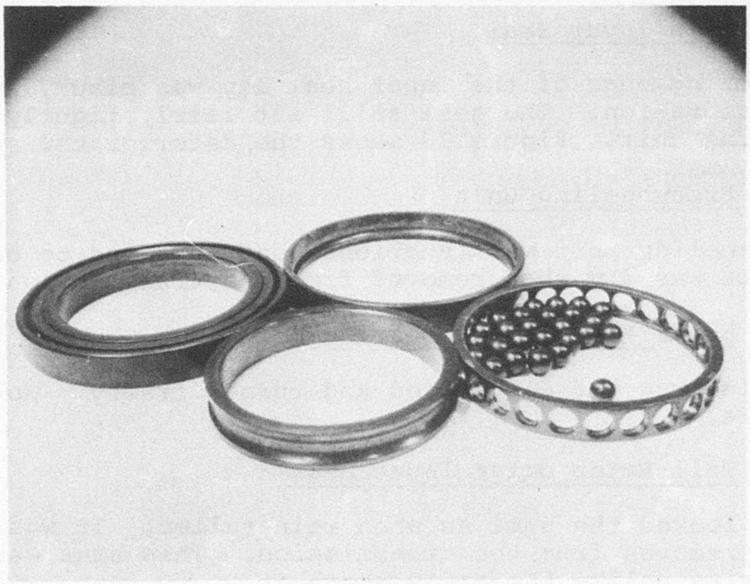

Figure 72. Input gearshaft duplex bearing following 4.0-hour emergency lubrication run.

5.4.5.11 Main Input Gear

The main input gear, which is shown in Figure 73, was scored slightly but was still functional. There was no indication of loss of backlash and no indication of the duplex creeping.

5.4.5.12 Main Input Pinion

The pinion shown in Figure 74 was slightly scored but was still functional. There was no indication of loss of backlash and no indication of the triplex creeping.

5.4.5.13 Main Input Triplex

The balls of the triplex bearing were discolored, but otherwise the bearing appeared to be unharmed, as can be seen in Figure 75. The inner and outer races and the silver-plated steel cages looked good.

5.4.5.14 Main Input Pinion Roller Bearing

The rollers of this bearing were discolored, and the bearing did not turn as freely as when installed. However, the rollers and races appeared to be in good condition and the bearing was still functional.

5.4.5.15 Main Input Seal

The contacting edge of the input seal lip was black, indicating seal deterioration. The seal still fit fairly tightly on the freewheeling unit. Figure 76 shows the deteriorated seal.

5.4.5.16 Freewheeling Unit

The freewheeling unit was functional and appeared to be unharmed. It was dry when removed from the transmission.

5.4.5.17 Lower Mast Roller Bearing

This bearing was not discolored and turned freely. No damage was visible.

5.4.5.18 Tail Rotor Drive Input Gear

This gear looked the same as when reinstalled. It was still oily when removed from the transmission. This area was free of coke deposits unlike the areas above it.

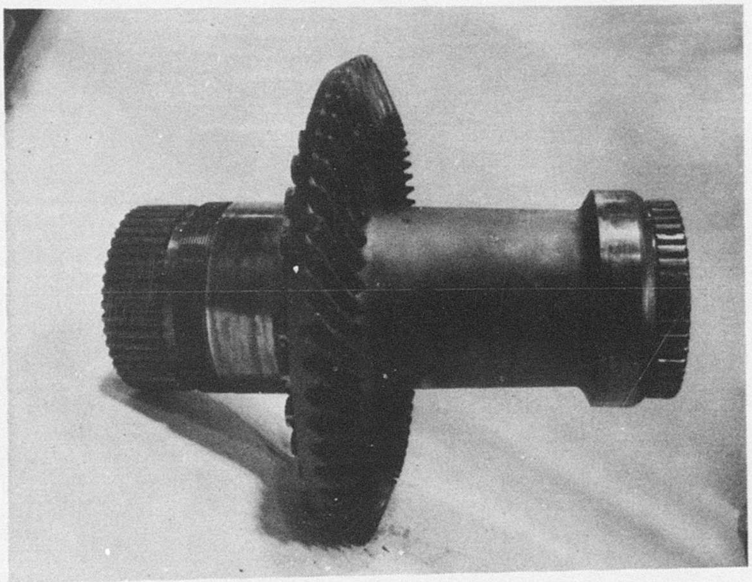

Figure 73. Main input gear following 4.0-hour emergency lubrication run.

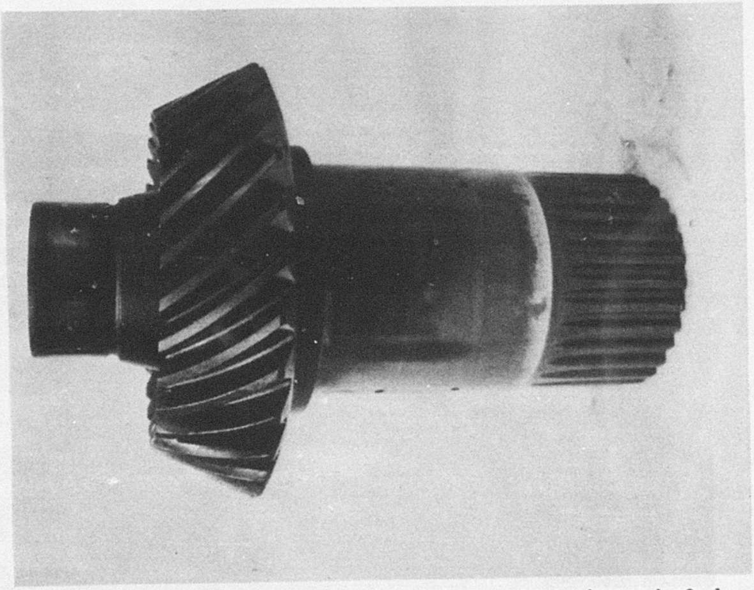

Figure 74. Main input pinion following 4.0-hour emergency lubrication run.

Figure 75. Main input triplex bearing following 4.0-hour emergency lubrication run.

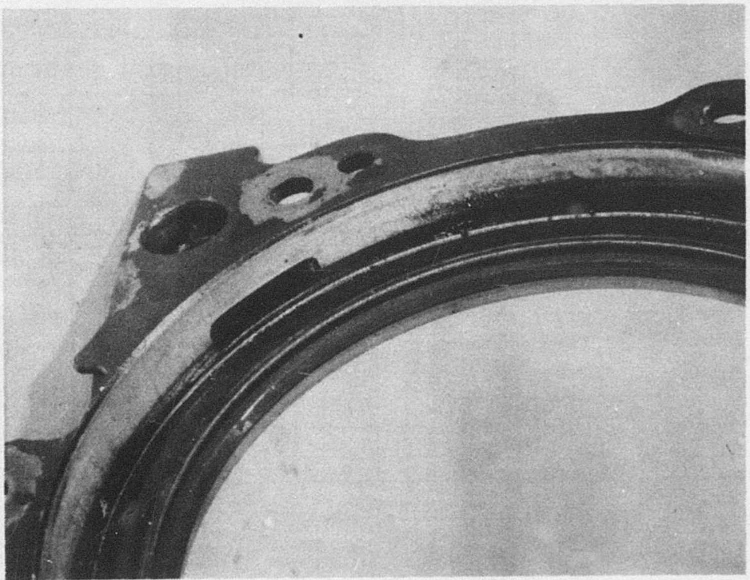

Figure 76. Main input seal following 4.0-hour emergency lubrication run.

5.4.5.19 Tail Rotor Drive Output Gear

This gear looked the same as when it was reinstalled. It was still oily when removed from the transmission.

5.4.5.20 Tail Rotor Drive Bearings

The quill assemblies were not disassembled, but the gears turned freely and the bearings appeared to be unharmed. They were still oily when removed from the transmission.

5.4.5.21 Upper Sump Case

The upper sump case, which is shown in Figure 77 showed no effects of the high temperature operation. The sides of the case were clean and not coated with coke deposits as were the upper cases. The aluminum oil deflector skirt was still bonded securely to the case.

5.4.5.22 Composite Sump Case

The composite sump case, which can be seen in Figure 78, showed no effects of the high-temperature operation. No delamination was observed. The paper gasket which was installed between the upper and lower sump case was not even browned, indicating that temperatures in that area stayed relatively low. Both the main pump and the emergency pump were functional and appeared to be unharmed. 205 ml (.22 quart) of oil remained in the emergency sump, and 1185 ml (1.25 quart) remained in the main sump. This oil could not be pumped since the level of the oil was below the pickup screen in each sump.

5.4.5.23 Main Support Case

Much of the interior of the main support case was coated with coke deposits. The paint on the exterior was not discolored.

5.4.5.24 Main Center Case

The interior of the main case was coated with coke deposits, as can be seen in Figure 79. Most of the paint on the exterior of the case was burned and discolored.

5.4.5.25 Ring Gear Case

Figure 80 shows the interior of the ring gear case, which was coated with coke deposits. The paint on the exterior of the case had burned off. Five nuts on the ring gear case lower flange and two nuts on the ring gear case upper flange had split and dropped off the bolts.

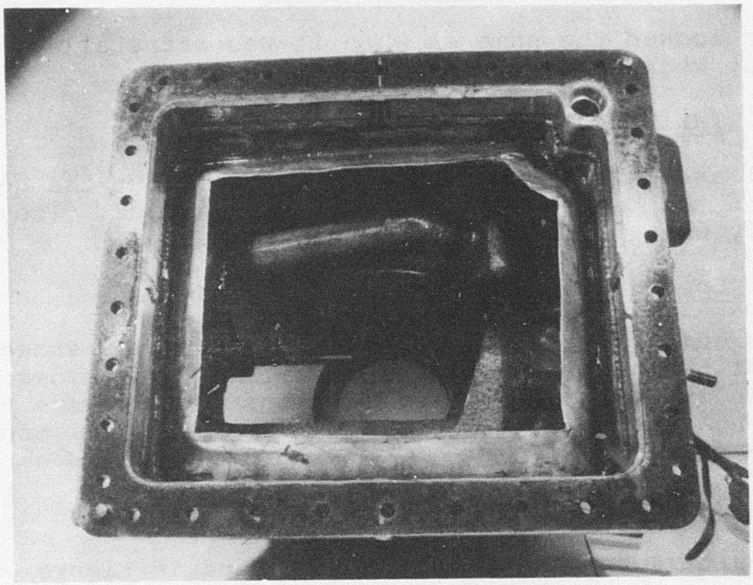

Figure 77. Upper sump case following 4.0-hour emergency lubrication run.

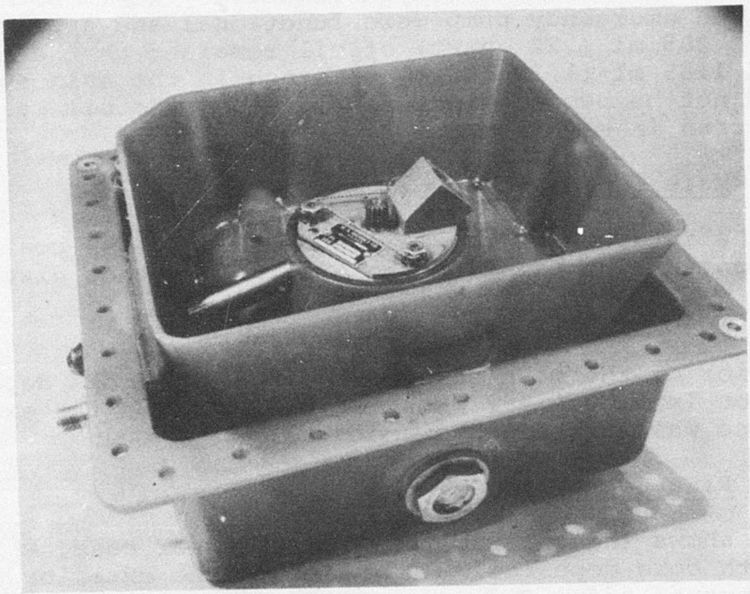

Figure 78. Composite sump case following 4.0-hour emergency lubrication run.

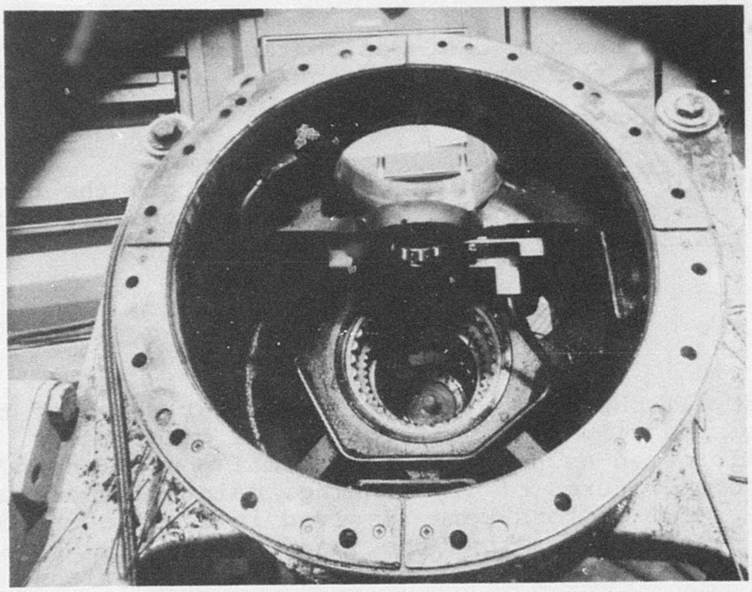

Figure 79. Main center case following 4.0-hour emergency lubrication run.

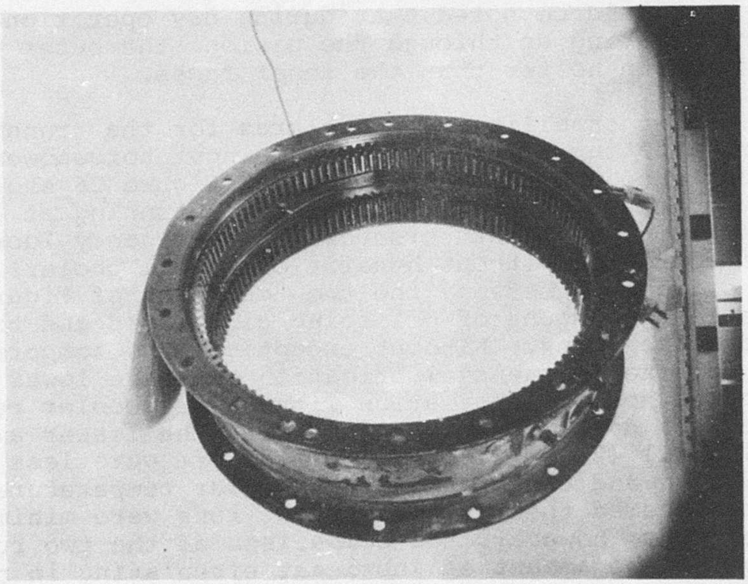

Figure 80. Ring gear case following 4.0-hour emergency lubrication run.

5.4.5.26 Top Case

The interior of the top case was coated with coke deposits, and the paint on the exterior was burned and discolored.

5.4.5.27 Mast

The portion of the mast which is housed within the transmission case was coated with coke deposits. Otherwise the mast appeared unharmed.

6.0 DISCUSSION OF RESULTS

Throughout all phases of testing, the outer races of the input triplex bearing were hotter than the inner races. This bearing performed as designed for normal operation and, at the elevated temperatures of dry running, maintained adequate clearance. Two factors were involved in causing the outer races to remain hotter than the inner races: the change in the outer race curvature, and the fact that lubricant was flowing up through the input pinion and through the inner races of the triplex into the bearing. The exact contribution of each of these factors to the success of the triplex bearing cannot be conclusively established without further testing; however, it should be noted that during dry operation when no oil was circulating up through the pinion, the outer races continued to run hotter than the inner races.

Figure 25 shows stabilized temperatures for the transmission running at 6600 input rpm and minimum input horsepower under normal lubrication with no oil cooler. Figure 26 shows stabilized temperatures for the transmission running at 6600 input rpm and minimum input horsepower on emergency lubrication (which of course is without benefit of an oil cooler). Thus, the only difference between the two test runs of Figures 25 and 26 is in the amount of oil being circulated and the number of oil jets operating. Without exception, the temperatures recorded for the emergency lubrication run were lower than those for the normal lubrication without oil cooler run. This, of course, was due to the fact that with the lesser amount of circulating oil (2 gpm versus 10 gpm) there were less losses due to windage and churning and thus lower temperatures. It must be remembered that these two test runs were minimum input horsepower runs; however, the comparison of the two runs implies that if the amount of lubricant circulating in normal operation could be reduced while still maintaining adequate lubrication, the efficiency of the transmission could be increased.

Figure 81 shows a plot of inlet oil temperature versus the oil temperature rise across the transmission (which is identical to the temperature drop across the oil cooler, ΔT) for 950 input horsepower and 1134 input horsepower with the standard mast installed in the transmission. Figure 23 shows a similar plot for 950 input horsepower with the stubbed mast installed. The plot of Figure 81 for 950 horsepower indicates that if the oil cooler were lost completely (for example due to ballistic damage), the transmission would stabilize at an oil-in temperature of approximately 325^{o}F. This utilizes the questionable data point of Phase II-1 step 4 discussed previously; however, this data point fits well the curve dictated by the other data points. Nevertheless, a stabilized oil-in temperature of 325^{o}F under normal lubrication with no cooler appears to be somewhat low since, as will be discussed later, under emergency lubrication in which there are less windage and churning losses as a result of less circulating oil, the recorded data indicate that the transmission oil-in temperature would stabilize at this same temperature of 325^{o}F. Test results of Phase I-3 step 1 and step 3 showed that at minimum input horsepower, there was 27^{o}F difference in the stabilized inlet oil temperatures of normal lubrication running with no cooler and emergency lubrication running. Still it would seem safe to estimate that under normal operation with no cooler at 950 input horsepower, the oil temperature would stabilize below 360^{o}F on a 100^{o}F day. In addition to this, extrapolation of the curve shown in Figure 32 indicates that it would take more than 1 hour to reach the stabilization temperature.

The 4-hour emergency lubrication test run of the high survivable transmission was divided into four distinct phases of operation, with each phase lasting approximately 1 hour. This can be seen clearly in the plot of the hottest monitored component in Figure 82. During the first phase (first hour) of operation, the transmission ran with the emergency lubrication system supplying sufficient oil to the gear teeth and rolling bearing contacts for both lubricity and cooling. The temperatures rose slowly and began to approach stabilization around 380^{o}F; then the emergency system began to run out of oil and the second phase (second hour) of operation started. The second phase appears to be a transition from lubricated and cooled contacts to just lubricated contacts. During this period, the vehicle for carrying the heat from the contacts to the case walls changed from oil to vapor, which dictated the search for a higher stabilization temperature. During the third phase (third hour) of operation, stabilization at approximately 475^{o}F was attained. During this period, the cases had become hot enough that they were now able to transfer virtually all of the heat being generated by the transmission. A small

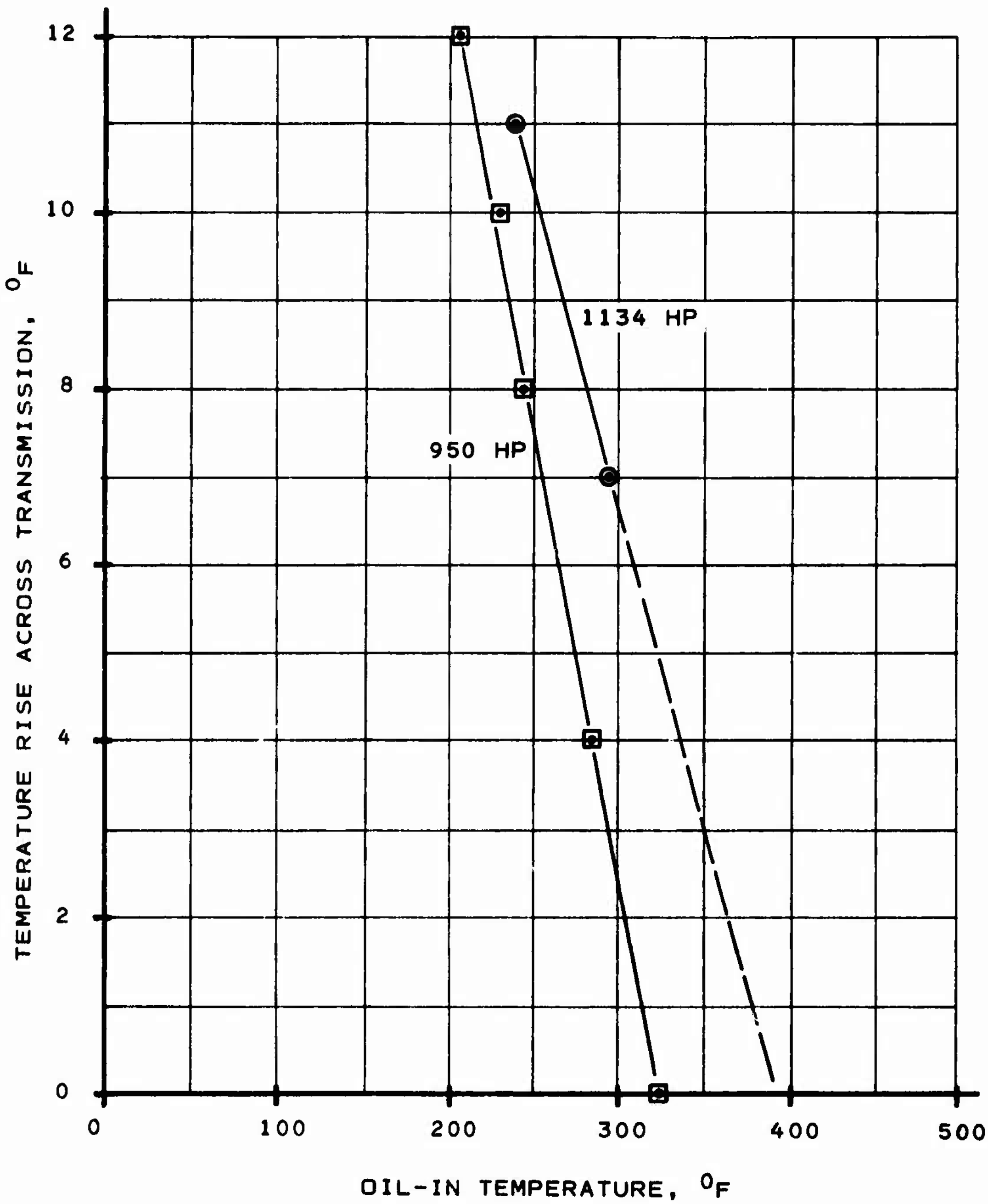

Figure 81. Oil cooler requirements of HST with standard mast at 950 horsepower and 1134 horsepower.

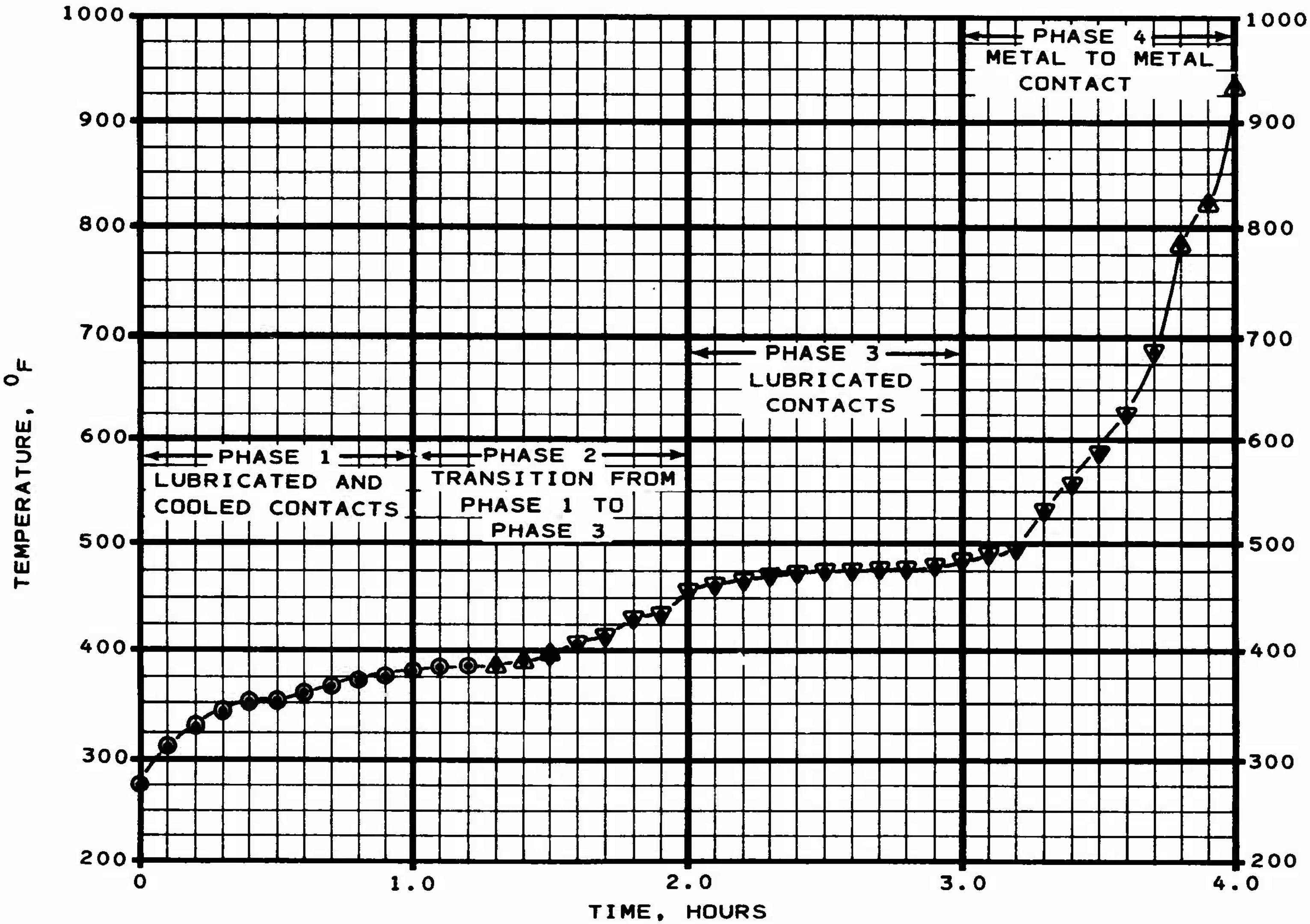

Figure 82. Plot of hottest monitored component during HST emergency lubrication test.

quantity of the generated heat was being dissipated through the evaporation of the oil that remained on the components. It is surmised that during this third phase, an oil film still existed on all the gear teeth and rolling bearing contacts to maintain a low coefficient of friction. It is further surmised that the heat generated by the transmission during the third phase was at a minimum for this power level, since there was only enough oil for lubricity and no excess available for churning losses and for increasing the internal atmospheric density to make the windage losses significant. Finally, the fourth phase (fourth hour) of operation was entered. During this phase, enough oil had evaporated that metal-to-metal contact resulted in some places, causing the coefficient of friction to increase. This led to much more heat generation which caused more oil evaporation and thus more metal-to-metal contact. Temperatures rose rapidly, and after exactly 4 hours of testing all torque was lost due to the gear teeth being stripped off the lower sun gear.

Two apparent conclusions may be drawn from Figures 40 and 82. The first is that the transmission could have run for an indefinite period of time at a stabilized oil temperature of approximately 325°F with the temperature of the hottest monitored component not exceeding about 400°F, had the emergency supply of oil not been lost. (It is presumed that most of the oil leaked out through the input seal.) This conclusion assumes that there would be no reduction in life due to the decrease in hardness and strength of the carburized gears, the 52100 bearings, and the magnesium cases and due to the degradation of the oil. The second apparent conclusion which may be drawn from Figures 40 and 82 is that at least 2 hours of operation (the third and fourth phases) after the loss of normal lubrication could have been achieved without the assistance of the emergency lubrication system. Before any firm decisions are based upon either of the two conclusions offered above, the following questions must be addressed:

1. Can the equilibrium condition attained at approximately 475°F in phase three be reached without the assistance of the emergency lubrication system unless the gear teeth clearances in both planetaries and the input bevel set are increased? The emergency lube system has been credited as having allowed all the components in the transmission to approach a temperature of approximately 400°F without any appreciable loss in clearances, thus allowing much higher temperatures to be reached before clearances were lost.

2. What effect, if any, did oiling the triplex bearing through slots in the inner races have on the difference in temperature between the outer and inner races? Calculations

show that the change in outer race curvature from 52% to 54% should force ball control to the inner races, thus generating less heat there than on the outer races. To confirm that this is definitely the case, this test would have to be run again without jetting any oil into the input bevel pinion for inner ring cooling of the triplex bearing.

3. Could similar results have been obtained using MIL-L-7808 oil rather than MIL-L-23699 oil? Since the evaporation rate at 475°F of 7808 oil is over three times that of 23699, it would seem that phases 2, 3, and 4 in Figure 82 would be considerably shortened if 7808 oil had been used.

4. How sensitive are phases 2, 3, and 4 to the power level at which the test is run? This and other tests have indicated that certain gearboxes can be operated for very long periods of time following the loss of oil, providing the power level is only about 0 to 25% of maximum continuous power. One of the emergency lube response runs during this test was made at 1134 horsepower (100% maximum continuous power). At the end of the 12-minute test run, the transmission oil temperature was 290°F, which was only about 10°F higher than that measured during similar response test runs at 950 horsepower (85% maximum continuous power). This indicates that the transmission can operate at 100% maximum continuous power indefinitely on the emergency lube system at a stabilized oil temperature of approximately 350°F. This assumes no reduction in life due to the decrease in hardness and strength of the carburized gears, the 52100 bearings, and the magnesium cases, due to the degradation of the oil. Although it appears very favorable, whether or not the transmission can indeed operate for 30 minutes after the loss of oil at 100% maximum continuous power could be determined only by test.

7.0 CONCLUSIONS

The HST test program led to the following conclusions.

1. The HST design is more than adequate to provide 60 minutes of operation following the loss of the normal lubrication system. The test results indicate that the transmission would have operated indefinitely at a stabilized oil temperature of approximately 325°F if the emergency oil had not leaked past the input seal.

2. In the event of the loss of the full effectiveness of the oil cooler (this could be due to loss of the blower), the HST would continue to operate on normal lubrication at a

somewhat higher stabilized oil temperature without serious transmission degradation. The tests show that if the oil cooler were capable of transferring enough heat to cool the incoming oil 4°F, the transmission could operate at 950 input horsepower at a stabilized oil temperature of approximately 300°F. In the event of total loss of the oil cooler effectiveness at 950 input horsepower, the oil temperature would rise slowly, and the test results indicate that after 60 minutes the oil temperature would still be below 350°F.

3. The 4-hour emergency lubrication run indicates that the HST could operate 60 minutes or longer after the loss of normal lubrication without the benefit of the emergency lubrication system, providing clearances were maintained in the planetaries and the input bevel gear set. The emergency lubrication system was credited with maintaining these clearances during the tests of this program. It is important to note that the transmission tested had no additional clearances ground in the planetaries or the input bevel set. The backlash on the input bevel set was the maximum blueprint allowable, but this was by chance and not by design.

4. The redesign of the input triplex bearing by changing the outer race curvature from 52% to 54% was the primary cause of an increase in internal clearance instead of a decrease during the test. Another factor that may have aided in maintaining cooler inner races than outer races was that the bearing was receiving lubricant through the input pinion. Just how effective the inner ring cooling was could not be determined during this test program.

5. The silver-plated steel retainers used in most all of the bearings for the HST performed very well even at the elevated temperatures.

6. The results of this test program imply that the amount of circulating oil within the transmission could be reduced significantly with a corresponding increase in transmission efficiency and with no detrimental effects.

8.0 RECOMMENDATIONS

The following recommendations are made for future applications of the concepts developed during this program.

1. A test program similar to the one conducted under this contract should be performed on two more transmissions assembled with the HST modifications to establish that the

success of the transmission tested here was not a random one. Future testing should be done using a radial carbon seal in the main input quill to prevent premature loss of the emergency oil supply.

2. Additional testing should be done to investigate whether or not 60 minutes of operation after the loss of normal lubrication could be achieved with a transmission modified with all of the HST modifications except the emergency lubrication system. Included in this investigation should be a determination of whether backlash in the planetaries and the input bevel gear set would need to be increased to achieve the 60-minute goal with no emergency lubrication.

3. Further testing of the modified input triplex bearing should be done to determine if the outer race curvature change, and not the lubricant being supplied through the inner race, was responsible for the excellent operating temperature characteristics exhibited by the bearing throughout this test program.

4. An investigation should be undertaken to determine whether or not the amount of circulating lubricant in the transmission could be reduced and still supply proper lubrication and cooling. Reducing the amount of circulating lubricant would result in less transmission losses due to windage and churning, and thus result in more efficient transmission operation.